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Zhang, Chen; Yu, Tao; Heiselberg, Per; Pomianowski, Michal Zbigniew

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Chen Zhang

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Aalborg University
Department of Civil Engineering
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by

Chen Zhang
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1. Introduction

The experiments are carried out in a climate chamber located at the Department of Civil Engineering Aalborg University. The objective of the experiments is to evaluate the performance of the system combining diffuse ceiling ventilation and thermally activated building construction (TABS) in terms of thermal comfort and energy performance. 20 cases with different boundary conditions are conducted varying on climate condition heat load. TABS water temperature and flow rate with or without diffuse ceiling. The energy evaluation includes energy balance of test room and cooling or heating capacity of TABS. And the thermal comfort is analyzed by draught rate vertical temperature gradient and radiant temperature asymmetry. Finally the effect of plenum and diffuse ceiling is discussed.

This report mainly focuses on the experiment results and discussions. Therefore, some details about the measurement are not presented in this report but can be found in the experimental plan made before.

2. Experiment description

2.1 The hotbox

The hotbox is divided into two chambers as shown in Figure 1 cold chamber simulates outdoor environment and hot chamber simulates a two-floor office building. The hot chamber is divided into three different zones. The lower zone (test room) represents a two-person office where the thermal comfort and energy performance are mainly investigated. And upper zone represents a second floor office which used to investigate the thermal behavior of TABS. It needs to notice that the concrete slabs are separated from the upper zone by means of a 50 mm insulation layer which aims to simulate the construction in the real building. There is always a layer of floor board or floor covering TABS reducing the heat exchange between TABS and upper zone and can prevent sound transmission. Lower and upper zones are enclosed by another zone which is named as surrounding zone. The set temperature in surrounding zone keeps identical to upper and lower zone to eliminate heat gain or heat loss from laboratory. In cold chamber, upper zone and surrounding zone air is re-circulated through the air handling unit (AHU) that can provide cooling and heating to individual zone depending on the predefined temperature profile.

Diffuse ceiling is installed at a height of 2.335 m. which consists of suspension profiles and acoustic ceiling panels. The acoustic ceiling panel is made of wood and cement which is known as cement-bounded wood wool panels. The measurements firstly run without diffuse ceiling and the last 10 cases are with diffuse ceiling.

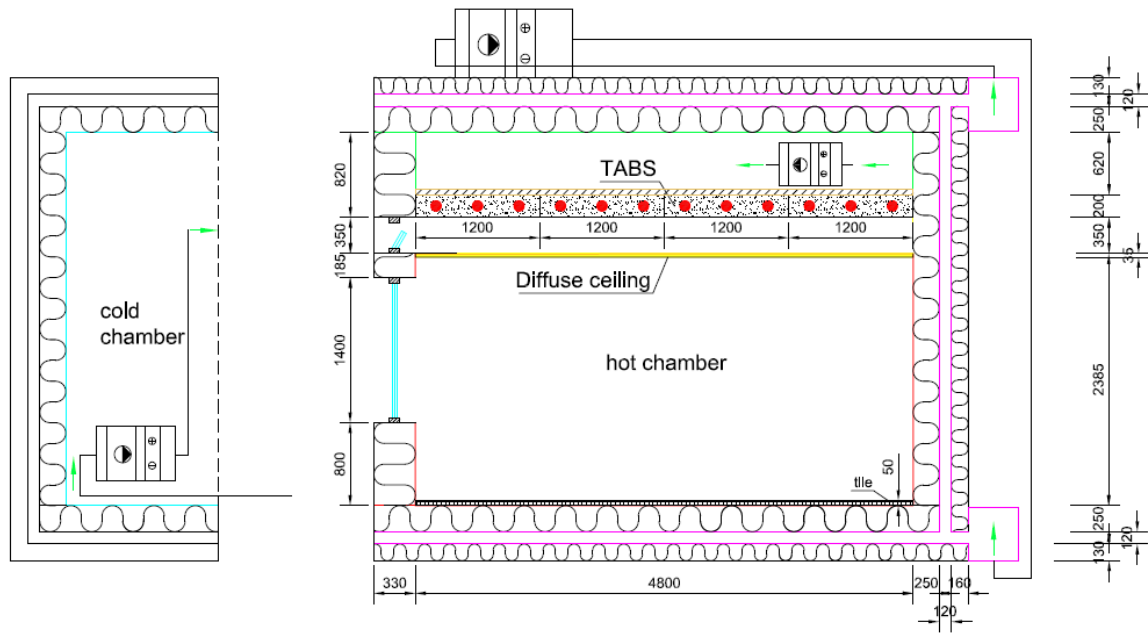


Figure 1 Vertical section view of hotbox (Unit: mm)

To simulate the real office environment, two workplaces are set up with two manikins, two computers on two tables in the middle of the room as well as two desk lights, as shown in Figure 2. The detailed power of the heat sources is given in Table 1, which is measured using the wattmeter before the test. Since the voltage in the lab is not so stable during the day and night, there is a small fluctuation of the power in the test and this will be included in the test error.

To simulate the conditions with and without solar radiation, two heat load levels-high heat load and low heat load are tested. For the low heat load level, only manikins, computers, desk lighting and monitors are used, corresponding to a level of 28.44 W/m^2 floor. For a high heat load, an additional electric carpet representing the solar radiation is located 30 cm close to the façade, corresponding to a level of 57.73 W/m^2 floor.



Figure 2 Distribution of heat sources in the test room

Table 1 Detail description of heat sources

Type	Power (W)	Type	Power (W)
Manikin 1		Manikin 2	
ventilator	33.0	ventilator	31.0
heating	67.0	heating	69.0
Computer 1	55.0	Desk lighting 1	54.0
Computer 2	45.0	Desk lighting 2	59.0
Monitor 1	21.5	Electric carpet	464.0
Monitor 2	16.0		
Type	Power (W)	Level (W/m ²)	
Low heat load	450.5	28.44	
High heat load	914.5	57.73	

2.2 Measuring points

The connection of TABS is in series and 5 water temperature measuring points are used for the thermal performance evaluation of TABS, which can be seen in Figure 3. These thermocouples measure the supply and return water temperature for each slab. At the middle of lower surface of each slab, there are three thermocouples for recording the slab lower surface temperature. Similarly, there is one thermocouple at the middle of each slab upper surface for the surface temperature.

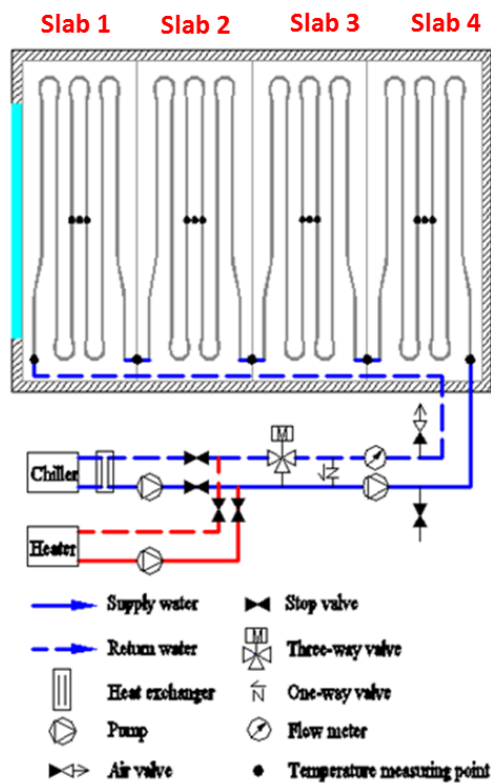


Figure 3 Hydraulic circuits of TABS

For the room temperatures, vertical temperature profiles are measured at 12 positions, as shown in Figure 4. But for cases with and without diffuse ceiling panel, the detailed measuring points are different and depicted in Figure 5. Since we only have 3 movable columns, during the test we have to move 4 times for the different positions. At each position, the measuring time is about two hours in order to get the stable conditions.

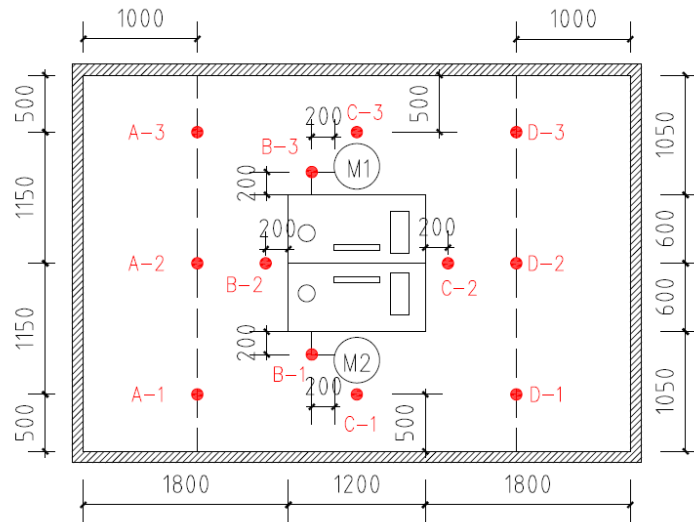
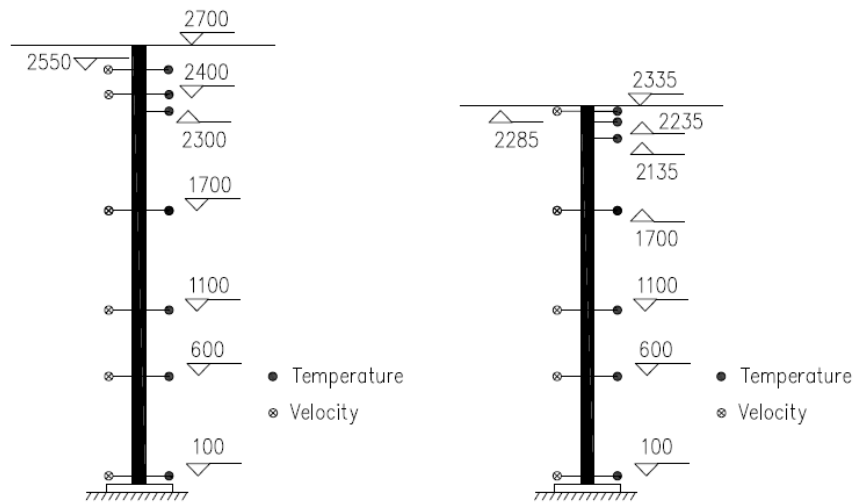
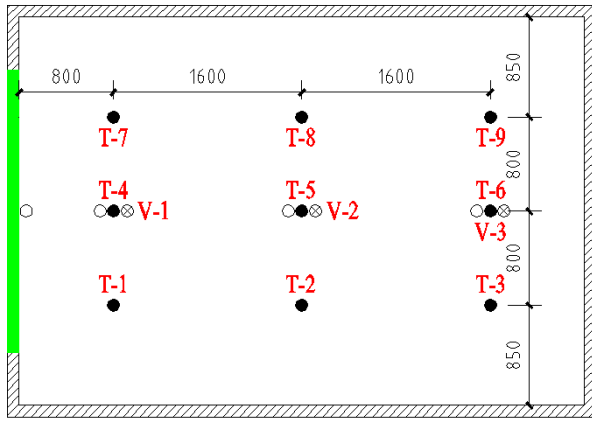


Figure 4 Measuring positions in the test room (Unit: mm)



(a) Without diffuse ceiling (b) with diffuse ceiling

Figure 5 Measuring points on the column (Unit: mm)



⊗ Velocity
 ● Temperature
 ○ Pressure

(a)

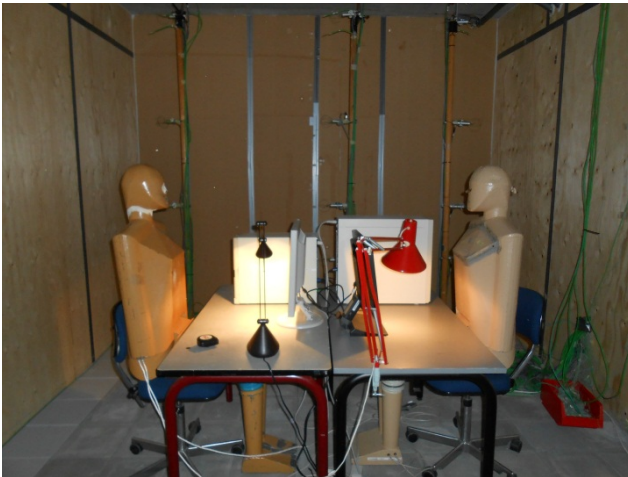


(b)

Figure 6 Measuring points in the plenum

In the plenum, air temperatures and velocities as well as the pressure difference are measured, the measuring points are shown in Figure 6. Since the height of plenum is very small, all measurements are carried out at the middle height of the plenum. Meanwhile, the surface temperatures of the ceiling panel are measured at the same location of air temperature measuring points. but at both surfaces of the ceiling panel. Therefore, totally there are 12 points for upper surface of ceiling panel and 12 points for lower surface of ceiling panel.

2.3 Designed cases



(a) without diffuse ceiling panel



(b) with diffuse ceiling panel

Figure 7 Set-up in the test room

The 20 designed cases are listed in Table 2, where 10 of them are without diffuse ceiling panel and the others are with diffuse ceiling panel. The set-up in the test room with and with diffuse ceiling

panel are shown in Figure 7.

Table 2 Designed cases

Heat load	ACH (h ⁻¹)	NV ('-' means heat loss. '+' means heat gain)	TABS ('-' means heat loss. '+' means heat gain)	Outdoor temperature (°C)	Indoor temperature (°C)	Solar heat gain (W/m ²)	With diffuse ceiling	Without diffuse ceiling
30 W/m ² (only internal heat gain)	2	-200%	+100%	-8	24	0	√	√
	2	-100%	0%	9	24	0	√	√
	2	-50%	-50%	15	24	0	√	√
	2	0%	-100%	24	24	0	√	√
	2	50%	-150%	32	24	0	√	√
60 W/m ² (internal and solar heat gain)	4	-200%	+100%	-8	24	30	√	√
	4	-100%	0%	9	24	30	√	√
	4	-50%	-50%	15	24	30	√	√
	4	0%	-100%	24	24	30	√	√
	4	50%	-150%	32	24	30	√	√

2.4 Response time measurement

The activation of thermally activated building systems (TABS) needs much time, depending on the construction of the deck and geometry of TABS system. The test of response time for TABS is carried out before the system performance measurement. A steady-unsteady-steady process is performed, and the time is recorded when 63% of the cooling energy is released.

In ISSO 85 [1], an example is given how to test the response time. In order to get an easier way to reach the steady state, the cooling case is considered in this study. The steps are listed as follows:

1. At the initial state, the temperature of water and surrounding zones at both sides keep at 25 °C to ensure a uniform temperature of 25 °C for TABS. The water flow rate of TABS is kept at 0.136 m³/h, corresponding to the velocity of 0.188 m/s in the pipes. (According to ISSO 85. water flow velocity should be between 0.15 and 0.30 m/s for optimal pressure loss and energy use)
2. At time=0 s. the water temperature in the pipes is suddenly changed from 25 °C to 15 °C and the water flow rate is kept the same value.
3. Both surrounding zones have the heating/cooling systems to keep the temperature of 25 °C, but due to the temperature difference of zone temperature and water temperature, cooling is achieved.
4. The time is counted until a stationary situation is achieved. i.e. max cooling capacity.

5. In this situation, response time is the time that is needed to achieve 63% of the end (max) value of cooling capacity.

To keep the indoor temperature of the test room with 25 °C, two radiators and one small fan are used. The radiators are used to provide enough heat loads and the small fan can make the room air more evenly distributed. Both the radiators and the fan are controlled by a regulator with an on-off control. A set-point of 25 °C is given, when the room air temperature is lower than 25 °C. the radiators and the fan are activated; when it is higher than 25 °C, they are switched off. The power of radiators and fan is recorded by the data acquisition system.

Normally, due to the time delay of concrete slab, the response time of water side and room side is different. The measurement is terminated when every recorded temperature reaches the finally stationary state. Finally, the cooling powers at the water side and at the room side are calculated.

3. Results and discussions

3.1 Response time for TABS

3.1.1 Cooling energy at water side

Figure 8 and Figure 9 show the results of cooling energy in the response time test. The cooling energy released by TABS can be calculated by the temperature difference between water inlet and outlet multiplying by water flow rate. As shown in Figure 9, the cooling energy by each slab is different. And it is not gradually decreased along with the water direction. This is because the heat transfer from the TABS depends on both radiation and convection. The radiation heat transfer is determined by surface temperature and the view factor to the other surfaces. And the convective heat transfer is determined by the temperature difference between surface and air as well as the convective heat transfer coefficient. As shown in Figure 3, the air inlet is located close to slab 1. While the air temperature increase and velocity decrease with the distance from the inlet, the convective heat transfer will also decrease with the distance from the inlet. This means that the performance of the TABS will be different if the water direction is changed and it can be discussed what direction gives the best result.

3.1.2 Cooling energy at the room side

The power used by the radiators and the fan is considered as the heat load removed by the TABS. The results can be found in Table 3. At steady-state, 619 W of heat is removed by TABS. The results show that the response time measured at the water side is 100 minutes, while the response time measured at the room side is 300 minutes. A time delay of 200 minutes can be found. This time buffer is caused by instead of directly release cooling energy into the room side, the cooling energy is firstly given to the concrete slab to cool down the surface temperature. After that the ceiling surface slowly releases cooling energy to the room by convection and radiation. Secondly is the heat resistance from water to concrete slab much smaller than the heat resistance between concrete slab and room.

Table 3 Response time test for TABS – calculated at water side

	Inlet temperature (°C)	Outlet temperature (°C)	Water flow rate (m ³ /h)	Start time (s)	Steady state-total cooling energy (W)	63% of total cooling energy (W)	Time of 63% cooling (s)	Response time (min)
4 slabs	15.13	19.51	0.136	64000	692	1016.8	70000	100.0
Slab 1	18.93	19.51	0.136	64400	91.5	125.4	91900	458.3
Slab 2	17.64	18.93	0.136	64330	204.0	279.5	65790	24.3
Slab 3	16.63	17.64	0.136	64230	160.4	219.7	79320	251.5
Slab 4	15.13	16.63	0.136	64120	238.6	326.9	70310	103.2

Table 4 Response time test for TABS – calculated at room side

Heat source	Start time (s)	Steady state-total cooling energy (W)	63% of total cooling energy (W)	Time of 63% cooling (s)	Response time (min)
Two radiators and one fan	64000	619	390	82000	300

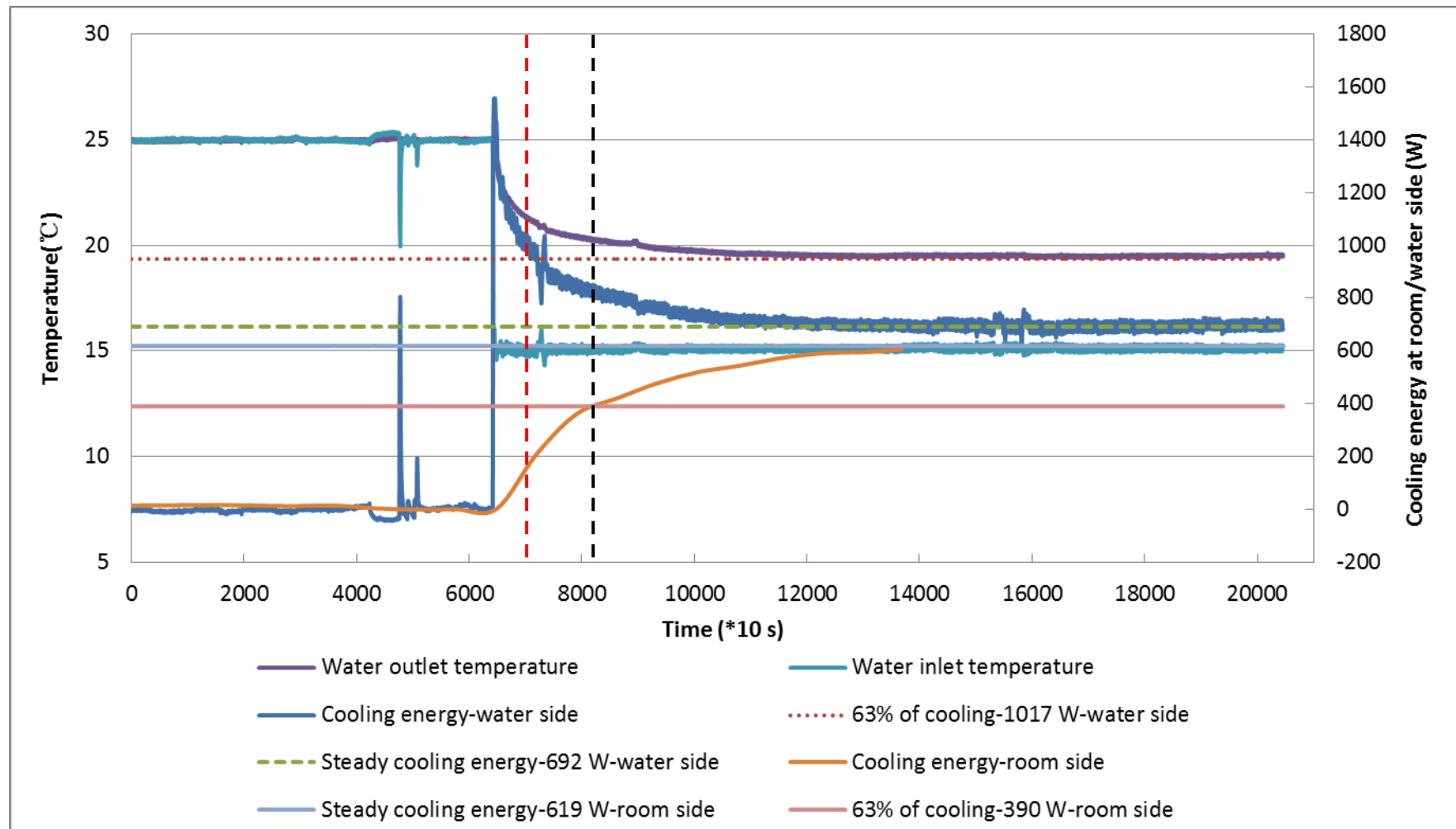


Figure 8 Total cooling energy results

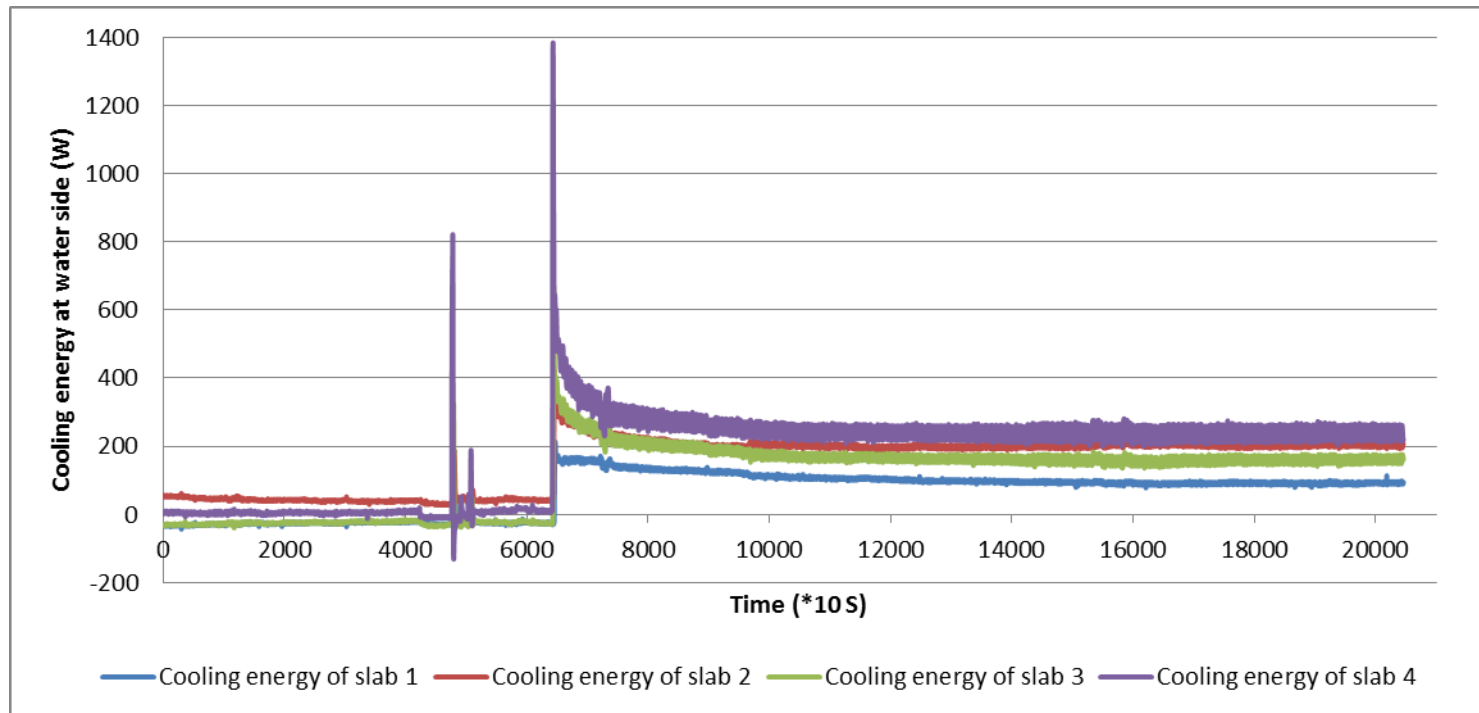


Figure 9 Cooling energy for each slab

3.2 Energy balances

For the evaluation of energy balance, we assume the heat gain in the room as positive value while the heat removed from the room as negative value.

3.2.1 Heat gain/loss from TABS to lower test room

For steady state conditions, the energy delivered by TABS can be calculated using the following expression:

$$Q_{TABS-tot} = C_{pw} \cdot M_w \cdot (t_{wsu} - t_{wre}) \quad (1)$$

Where:

$Q_{TABS-tot}$: Total energy delivered by TABS (W)

C_{pw} : Water specific heat capacity (J/kg K)

M_w : TABS water flow rate (kg/s)

t_{wsu} : TABS supply water temperature (°C)

t_{wre} : TABS return water temperature (°C)

The water flow rate is recorded from the water flow meter outside the test chamber, which is calibrated before the test. Since there is a little energy loss from the water pipes outside the test room, water temperatures are measured using thermocouples located in the water pipes inside the room.

To calculate the heat flow from TABS to the lower room, the heat loss to the upper zone should be deducted. Since the upper surface temperature of TABS is evenly distributed with the use of insulation. We can use the mean surface temperature of all slab upper surfaces to calculate the heat loss to the upper side.

$$Q_{TABS-up} = U_{insulation} \cdot A \cdot (t_{sur} - t_{au}) \quad (2)$$

Where:

$Q_{TABS-up}$: Heat loss of TABS to upper zone (W)

$U_{insulation}$: Heat transfer coefficient of the insulation and air film (W/m² K)

A : Floor area (m²)

t_{sur} : TABS upper surface mean temperature (°C)

t_{au} : Upper zone air temperature (°C)

Therefore, we can calculate the energy delivered by TABS to the lower test room:

$$Q_{TABS} = Q_{TABS-tot} - Q_{TABS-up} \quad (3)$$

3.2.2 Heat gain/loss from natural ventilation

The energy balance of natural ventilation can be calculated using the following equation:

$$Q_{NV} = C_{pa} \cdot \rho_a \cdot ACH \cdot V \cdot (t_{atm} - t_{asx}) \quad (4)$$

Where:

Q_{NV} : Energy delivered from natural ventilation into the room (W)

C_{pa} : Air specific heat capacity (J/kg K)

ρ_a : Air density (kg/m³)

ACH : Air change rate of natural ventilation (h⁻¹)

V : Room volume (m^3)

t_{air} : Inlet air temperature ($^{\circ}C$)

t_{exh} : Exhaust air temperature ($^{\circ}C$)

One thermocouple is located at the middle of each opening of the window respectively. We try to make the air in the cold box well-mixed. and the average value of these three inlet air temperatures is used for the calculation.

3.2.3 Heat gain/loss from constructions

The heat transmission from the constructions (facade, ceiling, floor, interior wall) can be calculated as:

$$Q_* = U_* \cdot A_* \cdot (t_* - t_{interior}) \quad (5)$$

Where:

Q_* : Heat transmission from the constructions (facade, ceiling, floor, interior wall) (W)

U_* : Heat transfer coefficient of the constructions and air film ($W/m^2 K$)

A_* : Area of the constructions (m^2)

t_* : Temperature of the zone corresponding to the constructions ($^{\circ}C$)

$t_{interior}$: Temperature of the interior surface of the construction ($^{\circ}C$)

After the construction of hotbox, the calibration work is carried out for both façade with and without windows. The calibration of U-value for the facade is shown in Table 5.

Table 5 Calibration of the facade

Only the wall	U without air film (W/m^2K)	U with air film (W/m^2K)
Case 1	0.16	0.15
Case 2	0.21	0.21
Case 3	0.21	0.2
Case 4	0.24	0.24
Case 5	0.20	0.19
Average	0.20	0.20
Wall with window		
Case 1	0.76	0.67
Case 2	0.67	0.59
Case 3	0.71	0.62
Average	0.71	0.63

When the small window is open for natural ventilation in the test, the U-value of the left wall and big window can be obtained based on the area. Finally, we can get this value is $0.61 W/m^2 K$. which is used for the heat transmission from the facade in the test.

For the ceiling slab, when TABS is not activated, the heat transmission should be calculated. For cases with ceiling panel, air temperature in the plenum and in the lower room will have different values. Therefore, interior walls will be divided into the lower room part and plenum part for the heat transmission calculation.

3.2.4 Total heat balance

The sensible heat balance of the test chamber can be calculated with the following equation:

$$\Delta Q = Q_s + Q_{NV} + Q_{TABS} + Q_{façade} + Q_{tw} + Q_{cell} + Q_{floor} \quad (6)$$

Where:

ΔQ : Error of the heat balance (W)

Q_s : Heat source power (W)

$Q_{façade}$: Heat transmission from the façade (W)

Q_{tw} : Heat transmission from the interior wall (W)

Q_{cell} : Heat transmission from the ceiling when TABS is un-activated (W)

Q_{floor} : Heat transmission from the floor (W)

The results of heat balance for all cases are listed in Table 6 and Table 7, the errors of heat balance (unbalance rate) are within 10% excluding cases 11 and 20. The error can be contributed to the data logger resolution, the composition of thermocouples, the accuracy of the ice point reference and the calibration as well as the fluctuation of voltage in the lab.

It can be seen from Table 6 that when there is no ceiling panel around 90 % of TABS cooling is supplied to the lower room, but TABS heating to the lower room only accounts for approximately 85 %. In Table 7 when there exists ceiling panel, more than 90 % of TABS heating goes to the lower room, but the proportion of TABS cooling is relatively low and increases with the ventilation air temperature and ventilation rate. The plenum is a closed zone for TABS, therefore, the thermal behavior of the plenum has a large influence on the energy performance of TABS. The thermal behavior of the plenum is largely dependent on ventilation rate and ventilation air temperature.

Table 6 Experimental results of cases without diffuse ceiling panel

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9	Case 10
Outside T (°C)	-7.10	9.21	15.79	23.82	31.63	-6.87	8.92	15.59	23.89	31.36
Inlet T (°C)	-7.58	9.71	16.47	24.19	31.76	-7.23	9.50	16.10	24.25	32.12
Exhaust T (°C)	22.17	23.57	24.12	24.66	24.67	20.16	23.65	23.64	24.07	24.55
Surrounding T (°C)	24.28	24.03	24.14	24.19	24.04	24.22	24.11	24.16	23.96	24.03
Upper T (°C)	24.75	24.24	24.10	24.18	24.11	24.62	24.11	24.24	24.10	24.13
Room air T (°C)	24.94	24.83	24.83	24.76	24.98	23.13	25.13	24.52	23.52	24.61
Operative T (°C)	24.15	24.54	24.54	24.52	24.54	22.66	25.18	24.48	23.33	24.20
Façade mean T (°C)	21.26	22.89	23.78	24.28	24.25	20.27	24.44	24.00	23.78	24.69
Left wall mean T (°C)	25.90	24.91	24.69	24.45	24.35	24.86	25.45	24.32	22.75	23.43
Inside wall mean T (°C)	26.27	24.77	24.29	23.99	23.42	25.07	25.07	23.64	21.96	22.32
Right wall mean T (°C)	25.89	24.70	24.69	24.47	24.18	24.87	25.41	24.24	22.79	23.23
Floor mean T (°C)	24.85	24.50	24.32	24.14	23.95	24.44	25.83	24.88	23.09	23.68
TABS inlet T (°C)	36.38	-	21.61	17.29	14.99	38.91	-	17.27	10.95	7.45
TABS outlet T (°C)	32.21	-	22.93	20.49	18.40	35.20	-	20.50	14.22	11.15
TABS T difference (°C)	4.17	-	1.32	3.21	3.40	-3.70	-	3.23	3.26	3.69
TABS water flow rate (m ³ /h)	134.50	-	143	140	200	208.00	-	141	273	337
ACH (h ⁻¹)	2	2	2	2	2	4	4	4	4	4
Heat load (W)	450.50	450.50	450.50	450.50	450.50	914.50	914.50	914.50	914.50	914.50
Facade heat loss(W)	-139.62	-67.35	-39.35	-2.28	36.34	-133.59	-76.37	-41.36	0.57	32.84
Ventilation heat loss(W)	-852.66	-397.02	-219.22	-13.40	203.01	- 1569.55	-810.76	-432.24	-19.97	433.91
TABS heat loss to test room (W)	556.45	-	-202.09	-472.89	-726.22	772.75	-	-478.35	-921.14	-1305.72
Interior wall+floor heat loss(W)	-14.70	-6.97	-3.63	-0.72	0.66	-6.01	-13.49	-1.16	13.31	8.77
Ceiling heat loss (W)	-	-7.62	-	-	-	-	-16.66	-	-	-
Heat unbalance (W)	-0.02	-28.47	-13.78	-38.79	-35.71	-21.90	-2.78	-38.61	17.68	84.31
Unbalance rate (%)	-0.01	-6.32	-3.06	-8.61	-7.93	-2.40	-0.30	-4.22	1.93	9.22

Proportion of heat removal										
Façade (%)	-30.99	14.95	8.73	0.51	8.07	14.61	8.35	4.52	0.06	3.59
Natural ventilation (%)	-189.27	88.13	48.66	2.97	45.06	171.63	88.66	47.27	1.14	47.45
TABS (%)	123.52	0.00	44.86	104.97	161.20	84.50	0.00	52.31	100.73	142.78
Proportion of TABS to test room (%)	85.41	-	92.20	90.60	91.76	86.10	-	90.43	89.00	90.27

Table 7 Experimental results of cases with diffuse ceiling panel

	Case 11	Case 12	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18	Case 19	Case 20
Outside T (°C)	-6.87	9.46	15.28	24.10	32.01	-7.23	9.41	15.32	24.07	31.99
Inlet T (°C)	-7.03	8.95	14.97	23.89	31.91	-7.82	8.49	14.88	24.18	32.06
Exhaust T (°C)	24.12	26.36	25.33	24.88	24.40	24.21	25.92	24.64	26.85	28.92
Surrounding T (°C)	24.15	24.17	23.96	24.13	23.97	24.04	24.05	23.93	23.98	23.89
Upper T (°C)	24.21	27.47	24.09	24.14	24.11	24.90	24.06	24.18	24.97	24.18
Plenum air T (°C)	20.09	17.56	16.10	15.19	14.59	10.84	12.42	12.62	15.82	19.56
Room air T (°C)	25.12	23.00	25.60	25.16	24.55	24.22	25.96	24.19	26.25	28.44
Operative T (°C)	24.49	26.48	25.29	24.80	24.22	24.12	25.87	23.98	26.14	28.10
Facade mean T (°C)	22.05	24.94	24.32	24.48	24.55	22.25	25.45	24.42	26.95	29.42
Left wall mean T (°C)	24.74	26.39	25.27	24.72	24.14	23.98	25.47	23.77	25.77	27.86
Inside wall mean T (°C)	24.67	26.07	24.85	24.16	23.50	23.53	24.64	22.80	24.75	26.77
Right wall mean T (°C)	24.65	26.10	24.92	24.49	23.95	23.46	25.10	23.31	25.36	27.52
Floor mean T (°C)	24.21	25.99	24.62	24.29	23.78	23.93	26.50	23.77	25.93	27.91
TABS inlet T (°C)	30.89	-	13.75	8.10	5.12	35.63	-	6.78	4.27	4.50
TABS outlet T (°C)	27.37	-	15.33	10.43	7.72	28.98	-	8.99	7.32	8.52
TABS T difference (°C)	-3.53	-	1.59	2.32	2.61	-6.65	-	2.22	3.04	4.02
TABS water flow rate (m ³ /h)	136.2	-	132.7	221.5	299.9	138.1	-	201.2	294.3	288.5

ACH (h ⁻¹)	2	2	2	2	2	4	4	4	4	4
Heat load (W)	450.50	450.50	450.50	450.50	450.50	914.50	914.50	914.50	914.50	914.50
Facade heat loss(W)	-135.91	-72.75	-42.51	-1.81	35.04	-138.58	-75.38	-42.74	-13.53	12.10
Ventilation heat loss(W)	-892.73	-499.00	-297.11	-28.31	215.34	- 1835.28	-998.83	-559.44	-152.99	179.81
TABS heat loss to test room (W)	518.62	-	-131.24	-429.95	-713.25	1011.44	-	-327.80	-810.43	- 1170.34
Interior wall+floor heat loss(W)	-3.79	-18.05	-8.73	-2.61	1.20	2.95	-12.71	4.81	-13.56	-33.31
Plenum wall heat loss (W)	1.82	4.14	6.37	8.13	8.58	6.11	8.16	10.16	8.54	5.69
Ceiling heat loss (W)	-	94.56	-	-	-	-	111.14	0.00	-	-
Heat unbalance (W)	-61.50	-40.59	-22.72	-4.04	-2.59	-38.86	-53.12	-0.50	-67.48	-91.55
Unbalance rate (%)	-13.65	-9.01	-5.04	-0.90	-0.58	-4.25	-5.81	-0.05	-7.38	-10.01
Proportion of heat removal										
Facade (%)	30.17	16.15	9.44	0.40	7.78	15.15	8.24	4.67	1.48	1.32
Natural ventilation (%)	198.16	110.77	65.95	6.28	47.80	200.69	109.22	61.17	16.73	19.66
TABS (%)	115.12	0.00	29.13	95.44	158.32	110.60	0.00	35.84	88.62	127.98
Proportion of TABS to test room (%)										
Proportion of TABS to test room (%)	92.5	-	53.6	71.9	78.4	94.7	-	63.2	79.5	87.9

3.3 Cooling capacity of TABS

3.3.1 Total cooling capacity

In order to assess TABS system performance, a model considering the cooling ceiling in steady state is used [6]. The A.U value of the TABS can be calculated as:

$$A.U = \frac{|Q_{TABS-tot}|}{\Delta T_{Ln}} \quad (7)$$

Where

$$\Delta T_{Ln} = \left| \frac{t_{wsu} - t_{wre}}{\ln(Arg)} \right| \quad (8)$$

$$Arg = \frac{t_{wsu} - t_{op}}{t_{wre} - t_{op}} \quad (9)$$

The mean value of operative temperatures measured at positions A-2, B-2, C-2 and D-2 at the height of 1.1 m is used for the calculation. This U-value includes the exchange with the test room and also the upper zone.

Similar as the heat exchanger, the ceiling effectiveness can be defined as follows:

$$Q_{TABS} = \varepsilon \cdot c_{pw} \cdot M_w \cdot (t_{wsu} - t_{op}) \quad (10)$$

$$\varepsilon = 1 - \exp(-NTU) \quad (11)$$

$$NTU = \frac{AU}{c_w} \quad (12)$$

$$c_w = c_{pw} \cdot M_w \quad (13)$$

Where

Arg : Temperature difference between the water side and room side

NTU : Number of transfer units

ε : Effectiveness of TABS

Top : Operative temperature (°C)

$A.U$: The heat transfer coefficient of TABS (W/K)

ΔT_{Ln} : Logarithmic mean temperature difference (K)

c_w : Thermal capacity of the water (J/K)

Using the above equations the system performance for cases without and with diffuse ceiling panel can be calculated as given in Table 8.

Table 8 Results of thermal performance of TABS ceiling cooling tests

	Case 3	Case 4	Case 5	Case 8	Case 9	Case 10	Case 13	Case 14	Case 15	Case 18	Case 19	Case 20
T_{wsu} (°C)	21.61	17.29	14.99	17.27	10.95	7.45	13.75	8.10	5.12	6.78	4.27	4.50
T_{wex} (°C)	22.93	20.49	18.40	20.50	14.22	11.15	15.33	10.43	7.72	8.99	7.32	8.52
M_{wsu} (kg/s)	0.0397	0.0389	0.0556	0.0392	0.0758	0.0936	0.03687	0.0615	0.0833	0.0559	0.08	0.0869
$Q_{cooling}$ (W)	219.17	521.98	791.43	528.98	1034.93	1446.49	244.73	597.95	909.55	518.41	1019.8	1462.6
Q_c (W/m ²)	13.84	32.95	49.96	33.40	65.34	91.32	15.45	37.75	57.42	32.73	64.38	92.34
Top (°C)	24.68	24.46	24.57	24.45	23.29	24.12	25.27	24.73	24.12	23.95	26.10	28.15
Arg	1.75	1.81	1.55	1.82	1.36	1.28	1.16	1.16	1.16	1.15	1.16	1.20
LMTD (°C)	2.35	5.42	7.75	5.40	10.63	14.74	10.71	15.44	17.73	16.04	20.27	21.57
A.U (W/K)	93.32	96.38	102.08	97.94	97.40	98.11	22.86	38.73	51.31	32.33	50.32	67.79
U (W/m ² .K)	5.89	6.08	6.44	6.18	6.15	6.19	1.44	2.45	3.24	2.04	3.18	4.28
NTU	0.56	0.59	0.44	0.60	0.31	0.25	0.15	0.15	0.15	0.14	0.15	0.19

ε	0.43	0.45	0.36	0.45	0.26	0.22	0.14	0.14	0.14	0.13	0.14	0.17
---------------	------	------	------	------	------	------	------	------	------	------	------	------

There is a linear growth of specific cooling power with the log mean temperature difference (LMTD) for cases without ceiling panel, as the black curve shown in Figure 10. While the cooling power of cases with ceiling panel does not increase linearly as the two curves depicted in Figure 10. Comparing the three curves in Figure 10, we can get the influence of ceiling panel on the cooling capacity of TABS. It is apparent that cooling capacity of TABS decreases due to the effect of ceiling panel at the same LMTD. That is to say, to meet a certain amount of cool demand, lower water temperature is indispensable for cases with ceiling panel. For the two curves of cases with ceiling panel, at the same LMTD the cooling capacity of TABS is higher for low heat load cases, this is because the plenum air temperature is obviously higher.

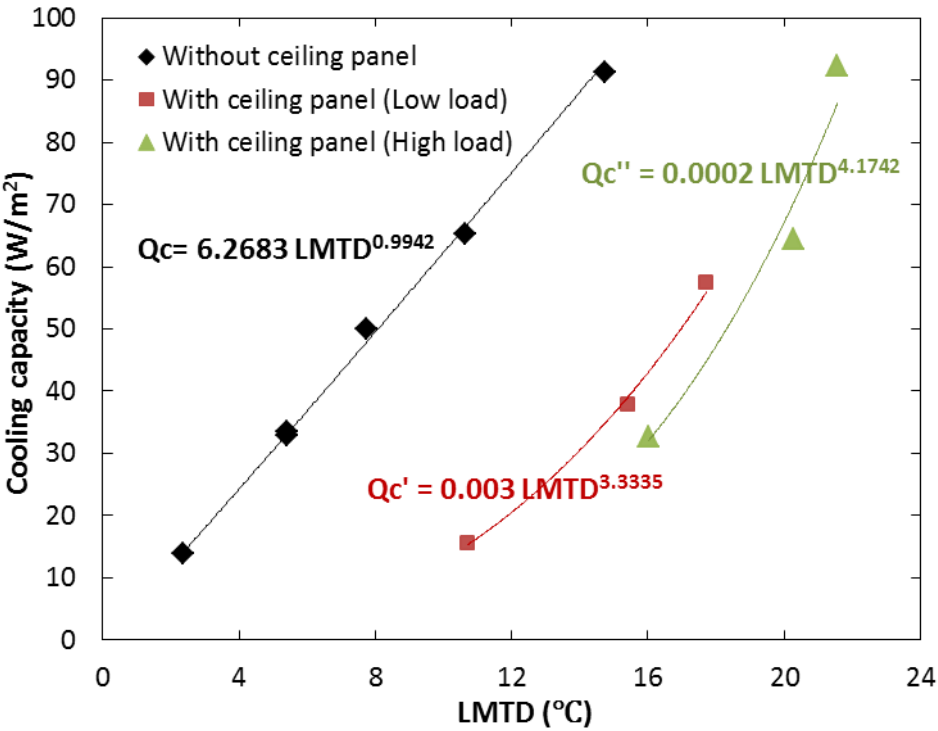


Figure 10 TABS cooling capacity as a function of the log mean temperature difference (LMTD)

Figure 11 gives the U-value of TABS cooling. It can be seen that it is almost a constant value of 6.2 W/m² K for cases without ceiling panel, while it almost linearly increases with LMTD for cases with ceiling panel. For the latter one, actually it is largely influenced by plenum air and ceiling panel.

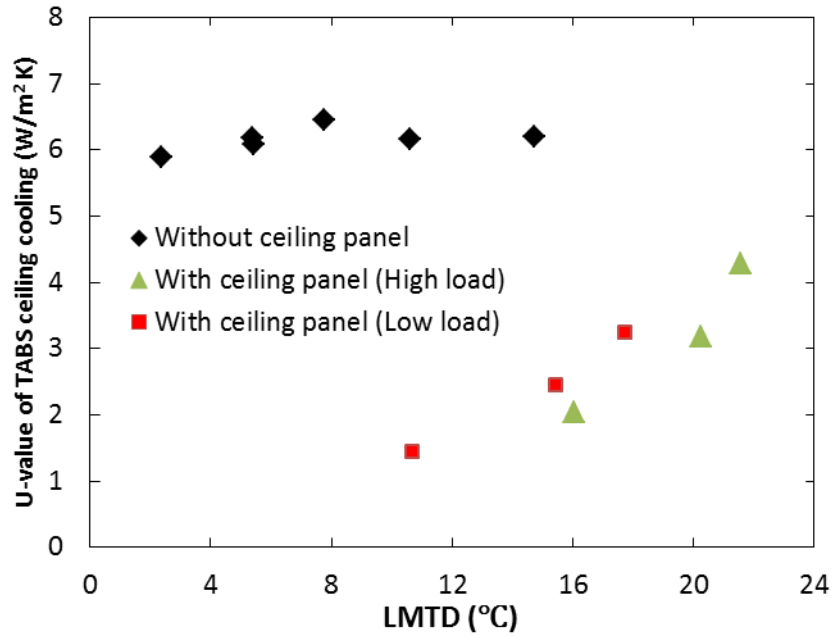


Figure 11 U-value of TABS variation with LMTD

Figure 12 shows the effectiveness of TABS. For cases without ceiling panel, ϵ decreases with LMTD. While for cases with ceiling panel, ϵ keeps almost the constant of 0.14. The reason for this low effectiveness is that the water temperature is extremely low in order to reach the sufficient cooling capacity.

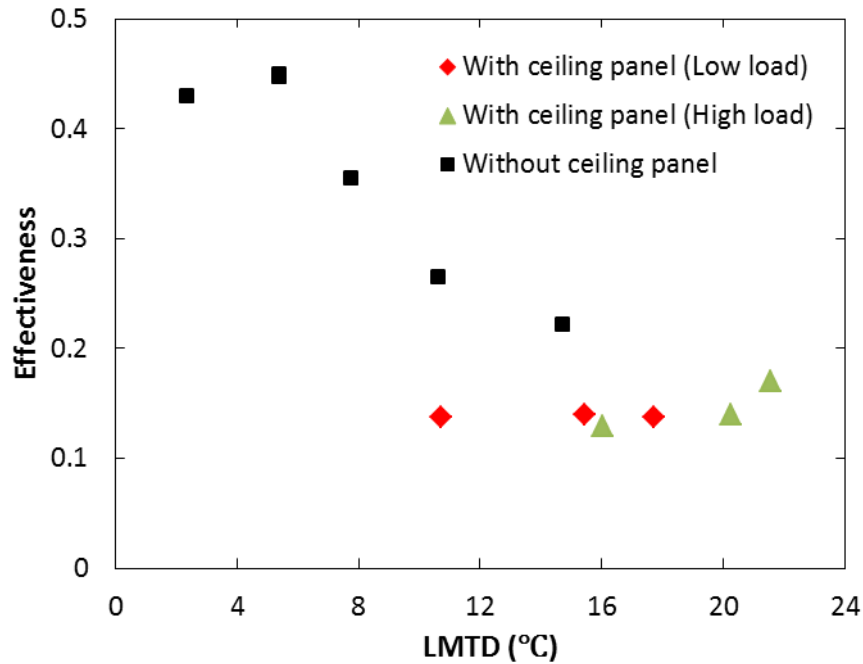
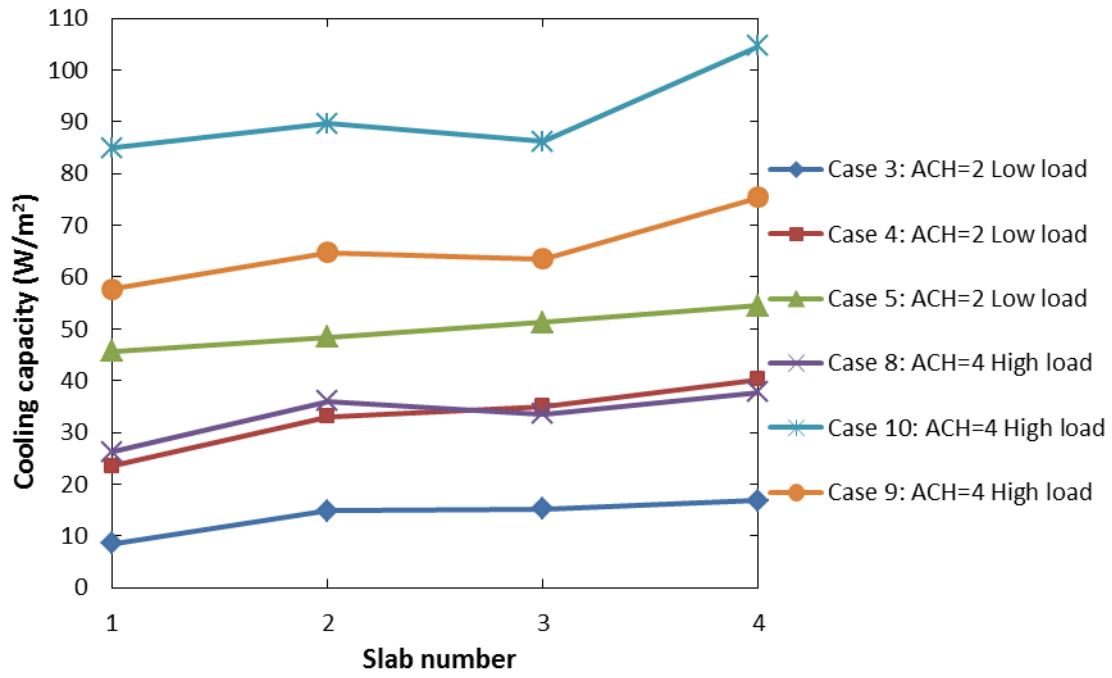


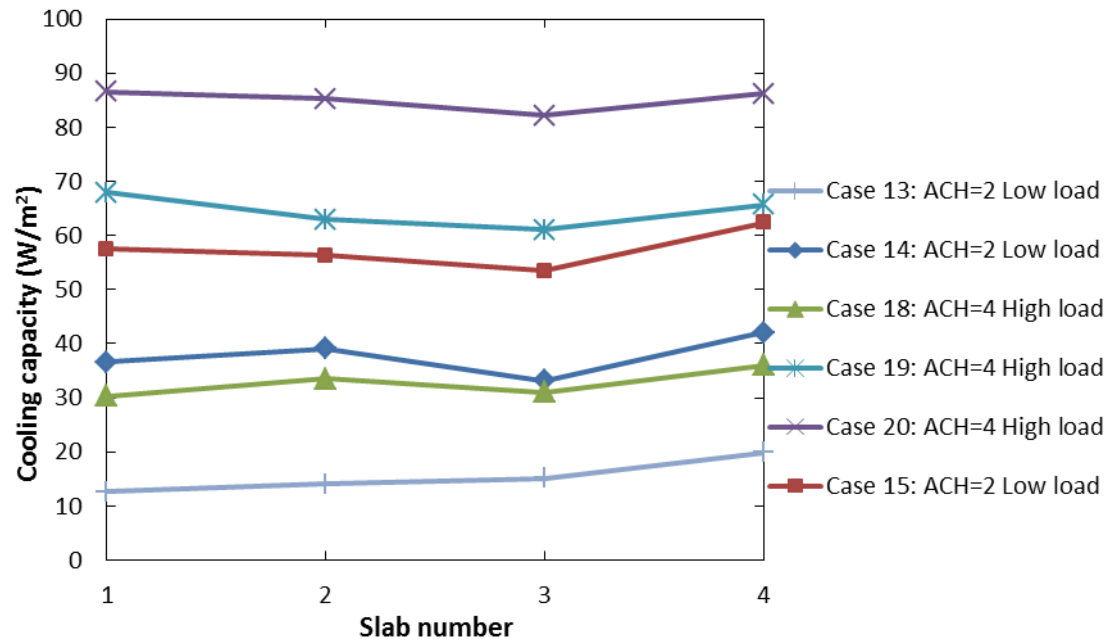
Figure 12 ϵ as a function of LMTD

3.3.2 Cooling capacity of each deck

In this test four slabs with TABS are used, since the water temperature increases for the cooling case, there is a need to investigate the cooling capacity of each slab. For different cases, many factors may influence the cooling capacity of each slab, such as the internal heat load level, air change rate (ACH), and so on.



(a) without ceiling panel



(b) with ceiling panel

Figure 13 Cooling capacity of each slab with TABS

Figure 13 shows the cooling capacity of each slab under different conditions. The slab of number 1 is the deck close to the façade, and the number 4 is the deck close to the inside interior wall or near the water inlet. The results show that the two slabs in the middle are more or less influenced the heat sources located at the middle of the room for cases without ceiling panel. However when there exists the ceiling panel, this influence is smaller.

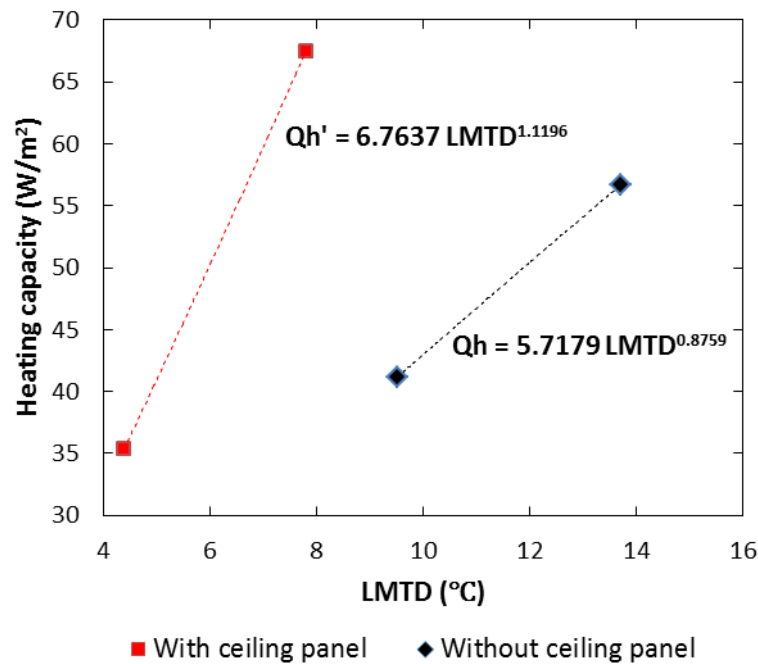
3.4 Heating capacity of TABS

The same evaluation method in Section 3.3.1 is used for the study of heating capacity. The system heating performance for cases without and with diffuse ceiling panel can be calculated as given in Table 9.

Table 9 Results of thermal performance of TABS ceiling heating tests

	Case 1	Case 6	Case 11	Case 16
Twsu (°C)	36.34	38.91	30.89	35.63
Twex (°C)	32.21	35.20	27.37	28.98
Mwsu (kg/s)	0.0374	0.0579	0.0380	0.0384
Qcooling (W)	651.52	897.93	560.84	1067.91
Qc (W/m ²)	41.13	56.69	35.41	67.42
Top (°C)	24.63	23.25	24.51	24.04
Arg	1.55	1.31	2.23	2.35
LMTD (°C)	9.51	13.72	4.39	7.80
A.U (W/K)	68.49	65.45	127.87	136.98
U (W/m ² .K)	4.32	4.13	8.07	8.65
NTU	0.44	0.27	0.80	0.85
ε	0.35	0.24	0.55	0.57

Since there are not sufficient heating cases measured, the heating capacity of TABS is not easy to determine. But we can still get the influence of ceiling panel on TABS heating, which is shown in Figure 14. It can be seen that the ceiling panel is beneficial to TABS heating, for the same heating capacity cases with ceiling panel need lower water temperatures.

**Figure 14** TABS heating capacity as a function of the log mean temperature difference (LMTD)

3.5 Thermal comfort performance

According to ISO 7730:2005 [2], draught risk, vertical temperature difference, radiant asymmetry are investigated to explore the local thermal discomfort.

3.5.1 Draught

Draught is one of the most common causes of complaint in ventilated or air conditioned buildings. It is defined as an unwanted local cooling of the body caused by air movement[3]. The discomfort due to draught may be expressed as the percentage of people predicted to be bothered by Draught. As state in ISO 7730, the draught risk should limit to 10% for Category A and 20% for Category B.

Calculate the draught rate (DR) using Equation:

$$DR = (34 - t_{a,l})(\bar{v}_{a,l} - 0.05)^{0.62}(0.37 \cdot \bar{v}_{a,l} \cdot T_u + 3.14) \quad (14)$$

Where

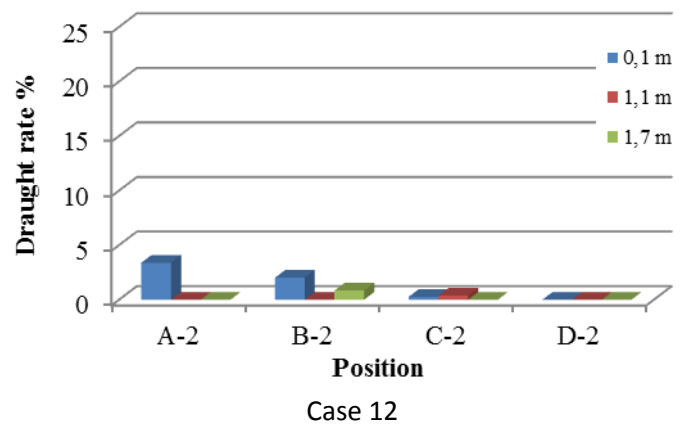
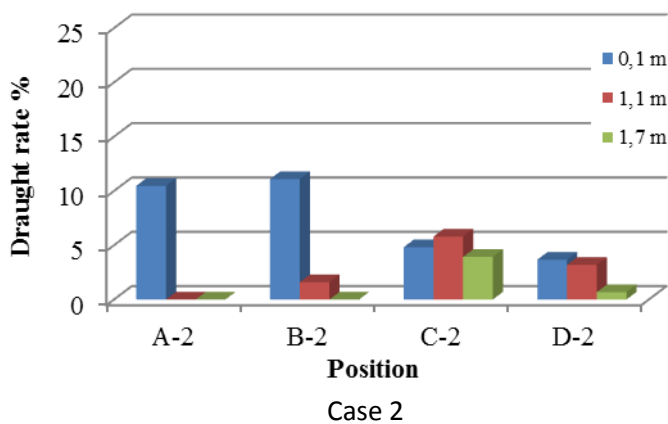
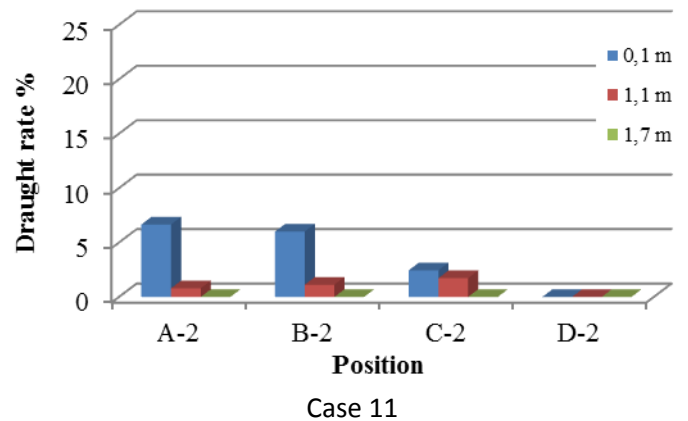
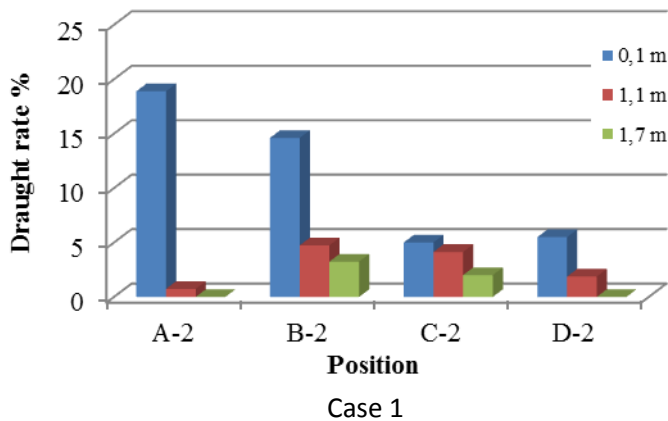
$t_{a,l}$ is the local air temperature, in degrees Celsius, 20 °C to 26 °C

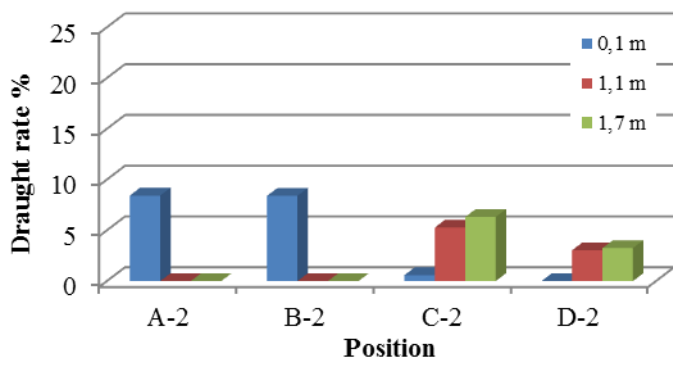
$\bar{v}_{a,l}$ is the local mean air velocity, in meters per second, < 0.5 m/s.

T_u is the local turbulence intensity, in percent, 10% to 60% (40% is used in this case)

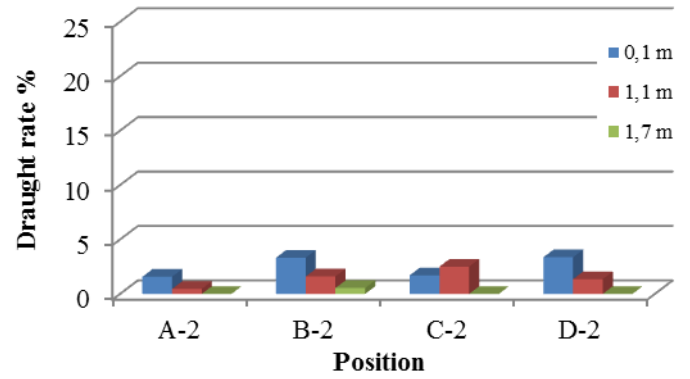
Draught rates are analyzed at 4 positions along the length of the room and 3 heights 0.1 m, 1.1 m and 1.7 m corresponding to ankle, head of sitting person and head of standing person, respectively. If compared the cases with and without diffuse ceiling at similar boundary conditions, it is obvious that low draught risk is experienced in the room installed diffuse ceiling. The biggest draught difference happens when the supply air temperature is low (Case 1 and Case 11, or Case 6 and Case 16). The highest draught risk in Case 6 takes place at the position closed to inlet with a DR value of 24%, while the draught rate is efficiently limited to 7% with the help of diffuse ceiling.

Second, from the view of horizontal distribution, the highest draught risk move from the place closed to façade to the deeper part of the room since the supply air temperature increase. This phenomenon happens in all the cases no matter with or without diffuse ceiling. This is because the supply air jet is significantly influenced by supply air temperature. The warm air can travel further along the ceiling, but the cold air will drop in a short distance from inlet due to gravity.

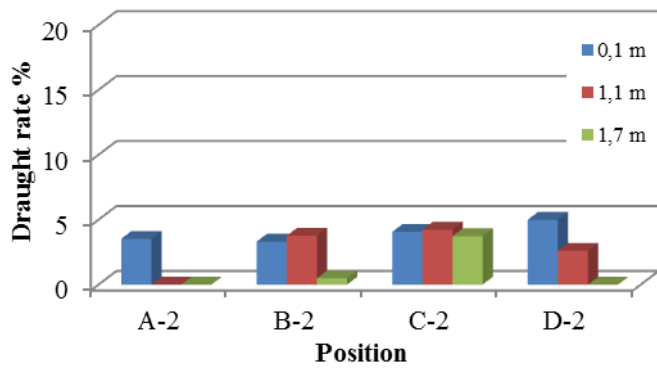




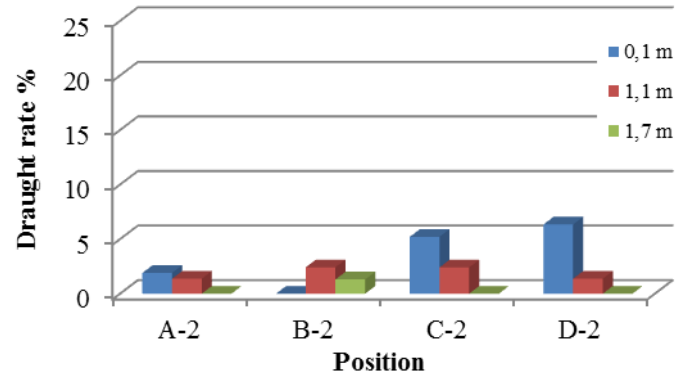
Case 3



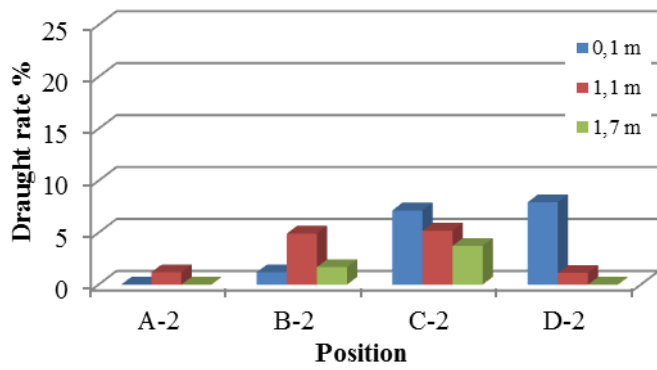
Case 13



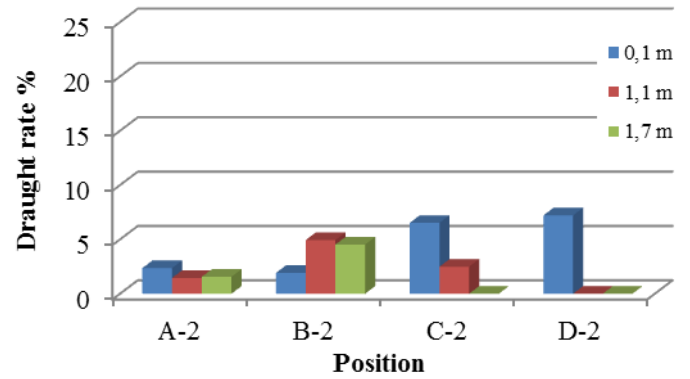
Case 4



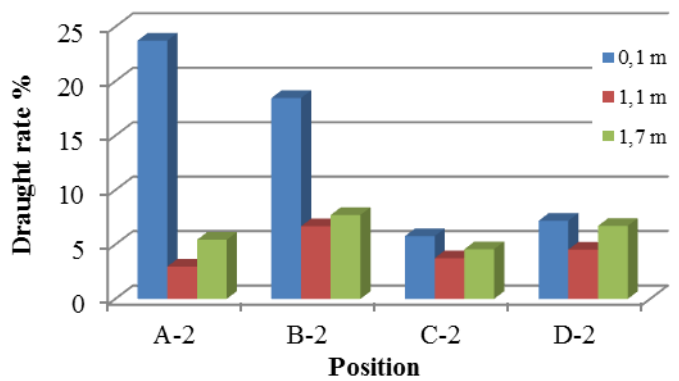
Case 14



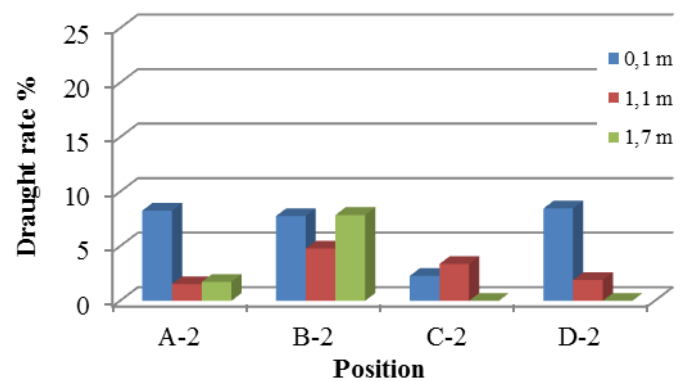
Case 5



Case 15



Case 6



Case 16

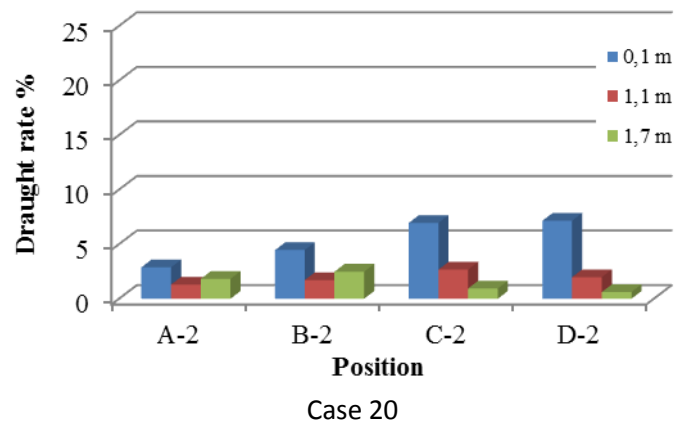
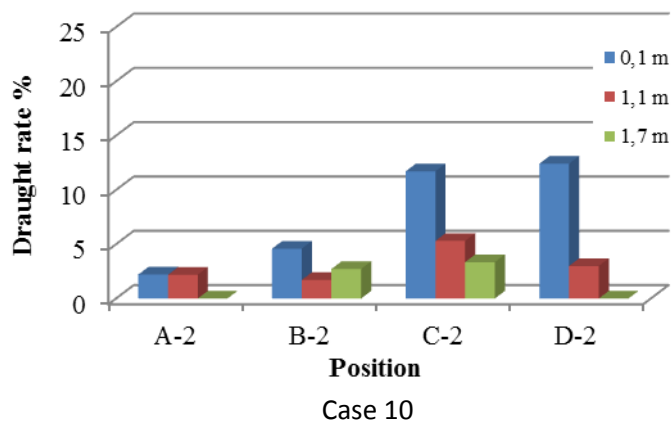
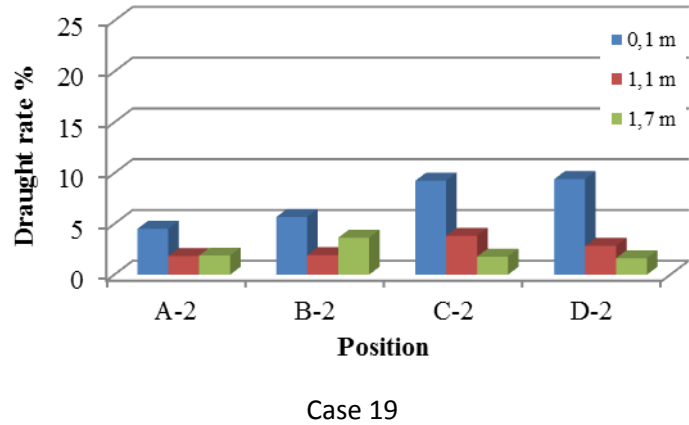
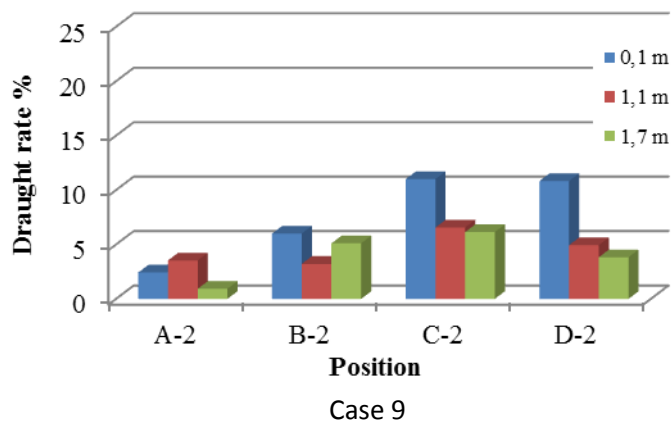
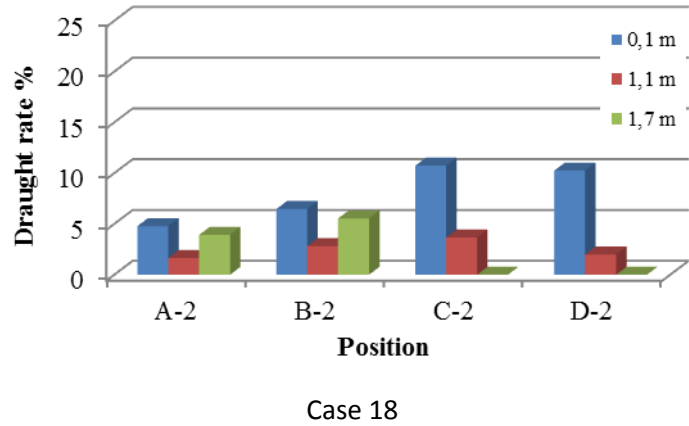
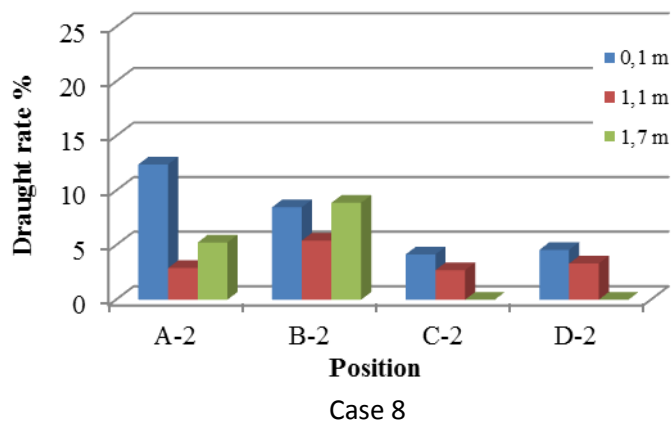
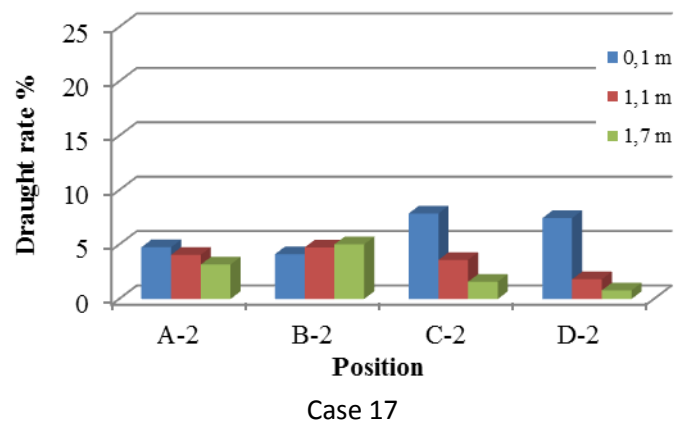
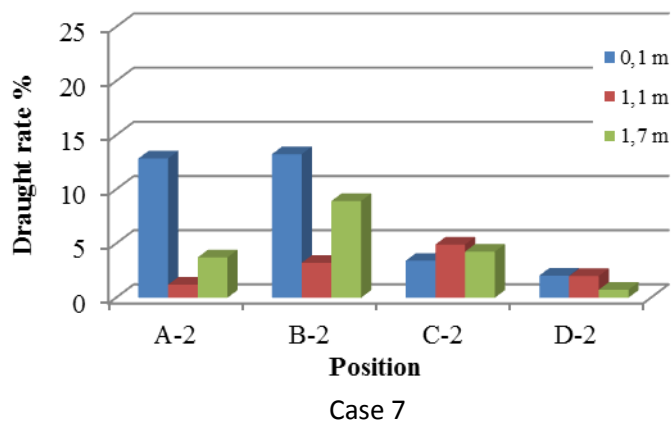
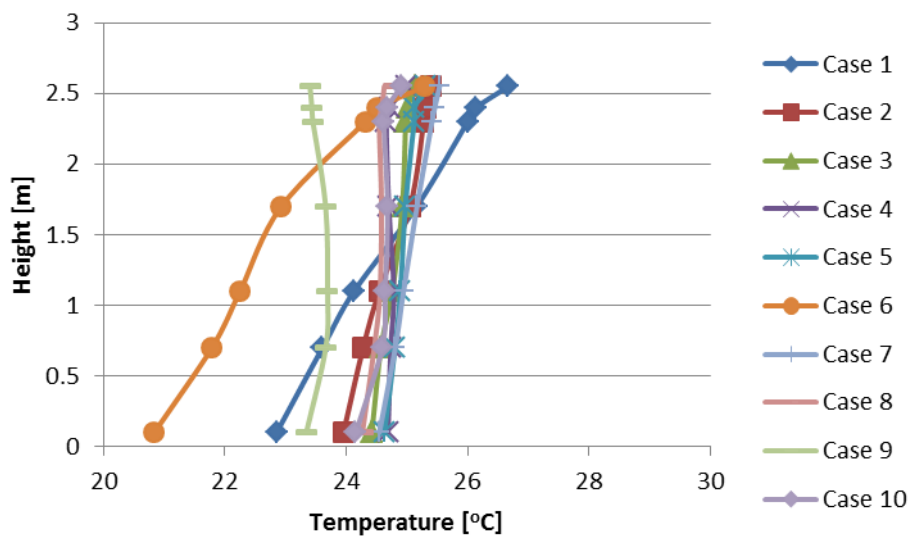


Figure 15 draught rates in each case

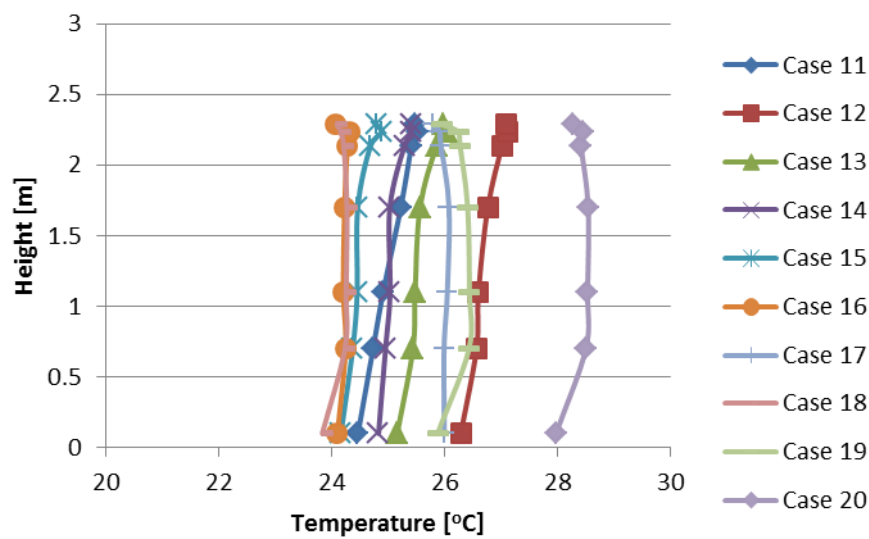
3.5.2 Vertical temperature gradient

Thermal stratification is an important problem for thermal comfort. High temperature between head and ankles can cause significantly discomfort. ISO 7730 specifies a maximum allowable air temperature difference of 3 °C between head (1.1 m for sitting person and 1.7 for standing person) and ankle (0.1 m).

Figure 16 shows the temperature gradient in the cases without diffuse ceiling and with diffuse ceiling separately. In the cases without diffuse ceiling, high vertical temperature gradient is found in Case 1 and Case 6 with low supply air temperature. The temperature gradient reaches 1.56 °C/m for case 1 and 1.82 °C /m for case 6. This means the cold air supply in to the room without a good mixing with the indoor air and cold air drop onto the floor directly which cause a low temperature at the floor level. However, the vertical temperature difference is significantly reduced with the help of diffuse ceiling. The temperature gradient is limited to 0.46 °C/m for all the cases with diffuse ceiling.



(a) Cases without diffuse ceiling



(b) Cases with diffuse ceiling

Figure 16 Vertical temperature gradient in the room

Table 10 Temperature gradient in each case

(a) Cases without diffuse ceiling

Case	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9	Case 10
Gradient (°C/m)	1.56	0.59	0.29	0.12	0.28	1.82	0.40	0.15	0.02	0.31

(b) Cases with diffuse ceiling

Case	Case 11	Case 12	Case 13	Case 14	Case 15	Case 16	Case 17	Case 18	Case 19	Case 20
Gradient (°C/m)	0.46	0.37	0.37	0.26	0.29	0.01	0.09	0.12	0.04	0.14

3.5.3 Radiant asymmetry

Differences in room enclosure temperatures can lead to thermal discomfort due to radiant asymmetry, even when the mean radiant temperature is within acceptable limits [4]. The radiant temperature asymmetry is estimated as the difference between the plane radiant temperatures in two opposite direction. As described by Fanger, it refers to a small plane element 0.6 m above the floor (the height of the center of a seated person). The small plane element should be horizontal to characterize radiant asymmetry caused by a warm or cooling ceiling.

People are less sensitive to radiant asymmetry caused by cool ceilings than warm ceilings or cool walls (windows). The radiant temperature asymmetry for warm ceiling should be less than 5 °C and for cool ceiling should be less than 14 °C to reach indoor environment category B[2]. Where the plane radiant temperature can be calculated by the equation below:

$$T_{pr}^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + \dots + T_N^4 F_{p-N} \quad (15)$$

Where:

T_{pr} is the plane radiant temperature, in Kelvins.

T_N is the surface temperature of surface N, in Kelvins.

F_{p-N} is the angle factor between a small plane element and surface N.

Table 11 Surface temperature of building envelopes and radiant temperature asymmetry between ceiling and other surface

(a) Cases without diffuse ceiling

Case	1	2	3	4	5	6	7	8	9	10
Facade	22.3	23.5	24.0	24.3	24.2	21.3	25.0	24.7	24.4	25.3
Inside wall	26.3	24.8	24.3	24.1	23.4	25.1	25.1	23.6	22.0	22.3
Left wall	25.9	24.9	24.7	24.5	24.4	24.9	25.4	24.3	22.7	23.4
Right wall	25.9	24.7	24.7	24.5	24.2	24.9	25.4	24.2	22.8	23.2
Floor	24.9	24.5	24.3	24.1	24.0	24.4	25.8	24.9	23.1	23.7
Ceiling (TABS lower surface)	32.1	24.9	22.9	20.6	19.1	33.8	25.5	20.5	15.8	13.9
Δt_{pr} [°C]	4.5	0.3	0.6	1.5	3.5	5.9	0.5	2.3	3.1	3.7

(b) Cases with diffuse ceiling

Case	11	12	13	14	15	16	17	18	19	20
Facade	22.0	24.9	24.3	24.5	24.6	22.3	25.4	24.4	26.9	29.4
Inside wall	24.7	26.1	24.8	24.2	23.5	23.5	24.6	22.8	24.7	26.8
Left wall	24.7	26.4	25.3	24.7	24.1	24.0	25.5	23.8	25.8	27.9
Right wall	24.7	26.3	25.1	24.5	23.9	23.5	25.1	23.3	25.4	27.5
Floor	24.2	26.0	24.6	24.3	23.8	23.9	26.5	23.8	25.9	27.9
Ceiling (Diffuse ceiling lower surface)	23.4	24.8	23.6	22.7	21.9	21.5	22.5	21.0	23.0	25.2
Δt_{pr} [°C]	0.6	0.7	0.5	0.9	0.9	1.6	2.8	1.7	1.8	1.6

Table 11 show the surface temperature of internal envelops and radiant temperature asymmetry caused by warm or cool ceiling. In the cases without diffuse ceiling, the ceiling surface facing to the occupants is the TABS lower surface. While, in the case with diffuse ceiling, diffuse ceiling lower surface serves as ceiling. It is clear to see that there is a risk of radiant asymmetry when TABS serves as heating system in the room without diffuse ceiling, where the temperature asymmetry of 4.5 °C closes to the limitation of 5 °C stated in ISO 7730. On the other hand, no evidence shows the risk of discomfort caused by the cold TABS surface, even though the TABS lower surface down to 13.9 °C. Even though no discomfort is observed in all cases, it is apparent that radiant asymmetry is reduced by installing diffuse ceiling in both heating and cooling cases. This is because instead of directly exposed heated or cooled TABS surface to the room, the TABS surfaces is encapsulated by diffuse ceiling and the radiation effect will be reduced simultaneously.

Actually, the issue limited the TABS surface temperature is not thermal comfort. it is the condensation risk. When the slab surface is lower than the air dew point, it will appear moisture condensation. In high humidity areas, the cool ceiling surface can dew condensation easily if the humidity is not controlled. Condensation will affect visual perception and function. and even drop water from the ceilings forming the so called ‘office rain’[5]. Figure 17 show the condensation on slab surface and diffuse ceiling surface, when the TABS surface temperature is 15.8 (Case 9) and 8.6 °C(Case 15).



(a)



(b)

Figure 17 Condensation on slab surface and diffuse ceiling surface

3.6 The effect of diffuse ceiling and plenum

The use of ceiling plenum to deliver air directly into the occupied zone is one of the key features that distinguish diffuse ceiling ventilation system from conventional ducted air distribution systems. Thermal processes within the plenum and the TABS have an important impact on the effectiveness

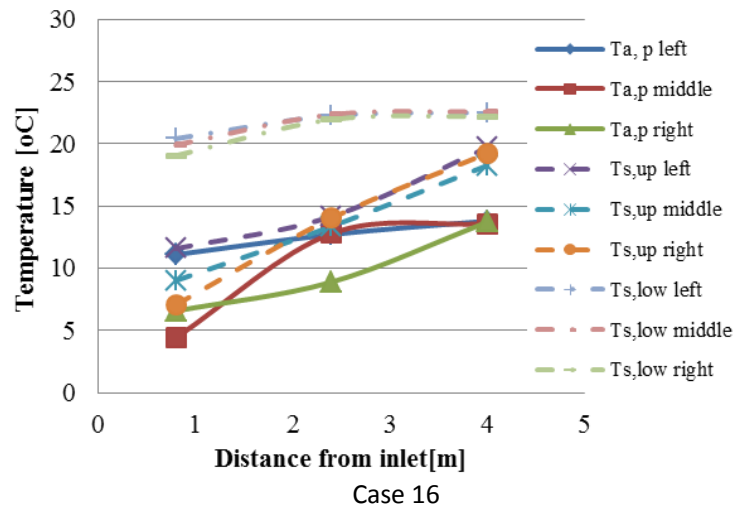
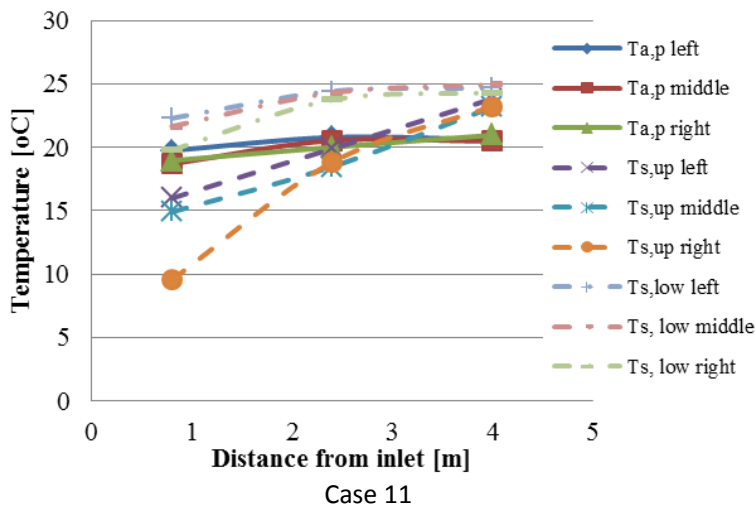
of TABS as cooling or heating system and plenum as air distribution system. The thermal processes include (1) heat transfer between the concrete slab and plenum air. (2) heat transfer between the diffuse ceiling panel and plenum air (3) heat transfer between concrete slab and diffuse ceiling panel (4) variations in plenum air temperature with distance traveled through the plenum

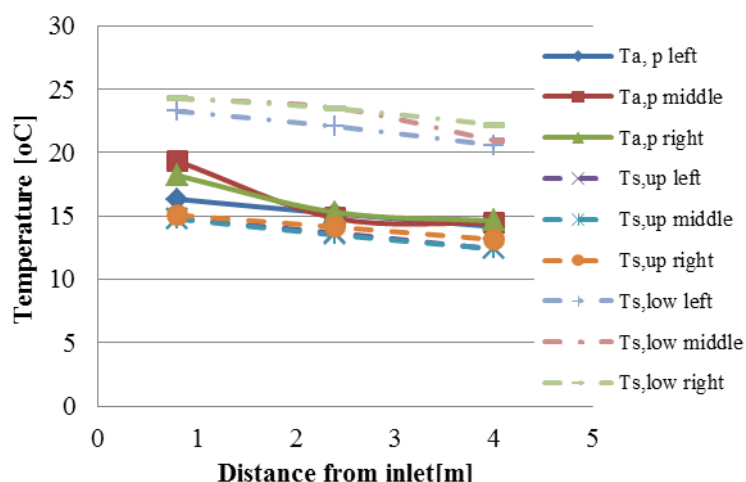
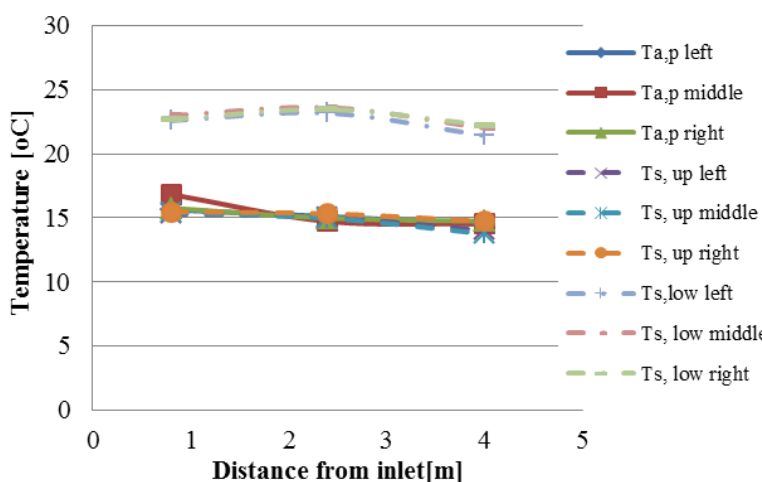
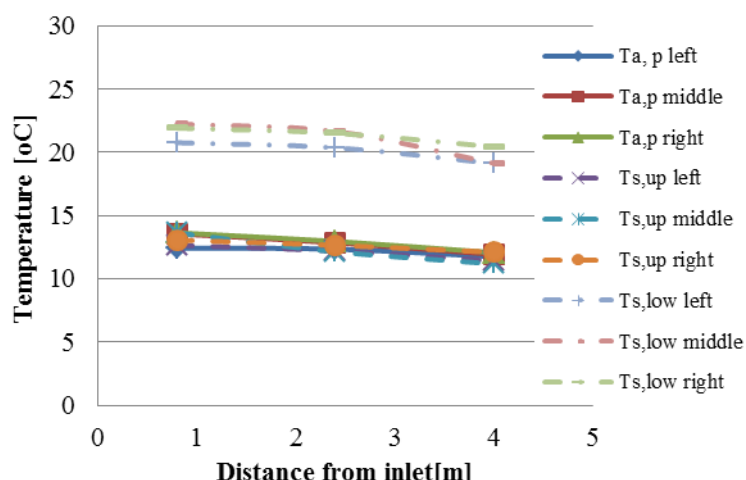
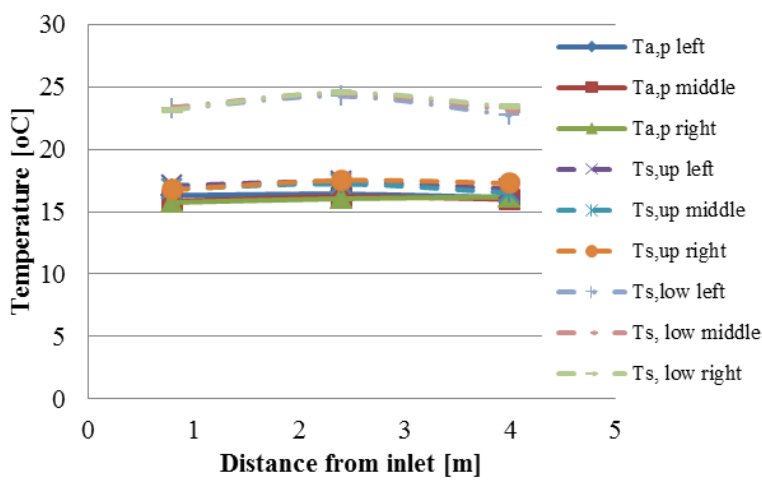
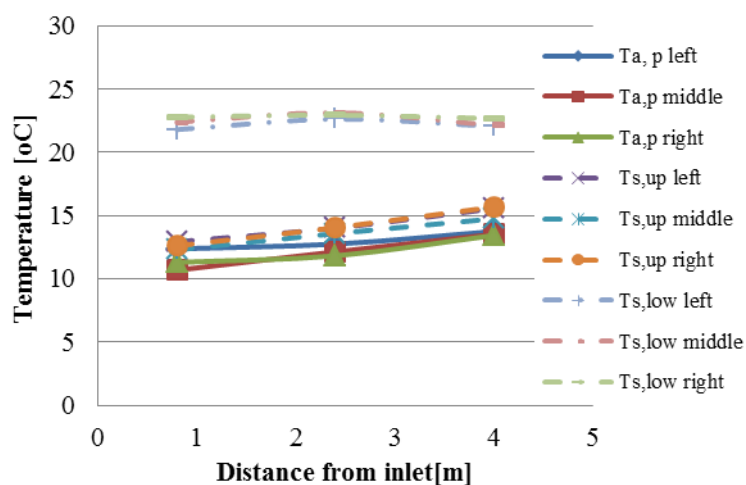
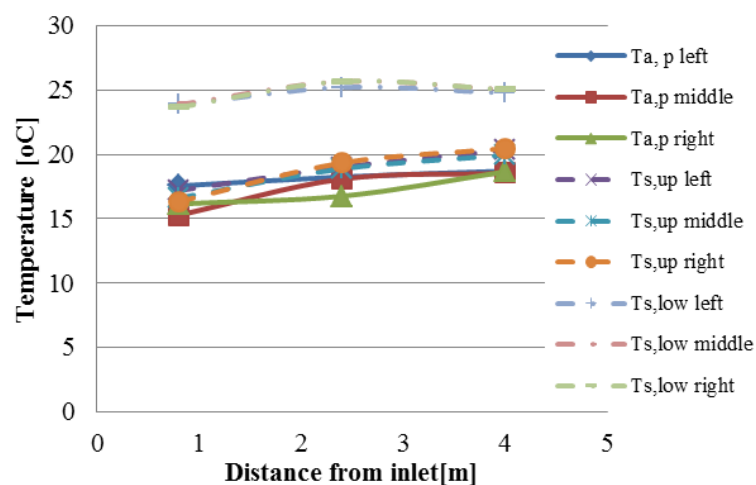
3.6.1 Temperature distribution in the plenum and diffuse ceiling surface

As shown in Figure 18, the air is pre-heat or pre-cool in the plenum before delivering into the occupied zone. When the air change rate is 2 /h (from Case 11 to Case 15), even though the supply air temperature range from -6.87 °C to 24. °C, the air temperature in the plenum remain relatively stable (average air temperature from 15.2 to 20.1 °C). The pre-heating or pre-cooling efficiency depend on the temperature difference between supply air and slab and diffuse ceiling surface. It is also influenced by the air flow rate. as indicated by Case 11 and Case 16.

On the other hand, it shows that the air temperature distribution in the plenum is as a function of the distance to inlet. In the cold climate, cool supply air enters the plenum through inlet on the façade. as it travels through the plenum it is warmed up by heat transfer from the concrete slab on the top (heat released from the TABS) and from the diffuse ceiling panel (heat conducted from the room). While in summer, warm supply air is cooled down by the same thermal process. However, from the design consideration, it is important to limit the amount of variation in supply air temperature to ensure the air supply into the occupied zone with relative uniform temperature.

A noticeable temperature difference between diffuse ceiling lower and upper surface can be observed in these cases, the temperature difference up to 10 °C. The lower surface has closed temperature to the room air and a peak surface temperature can be found corresponding to the middle of the room as the surface is heated by thermal plume generated by heat source, while, the upper surface has the similar temperature to the air in the plenum. Therefore, unlike the research performed on the perforated aluminum ceiling tile by Hviid [6]. no clear radiation cooling potential of diffuse ceiling is observed in our cases. This is due to the low conductivity of cement-wood ceiling tile used in our study, with λ -value of 0.085 W/m.K. while the conductivity of aluminum is up to 205 W/m.K.





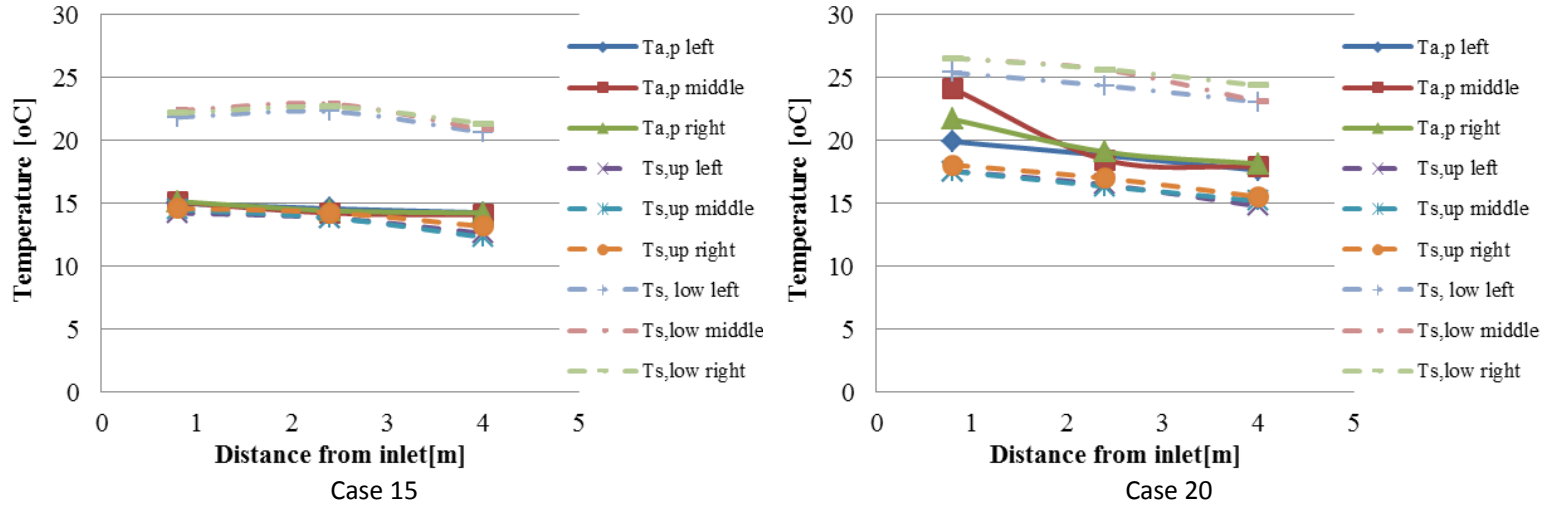
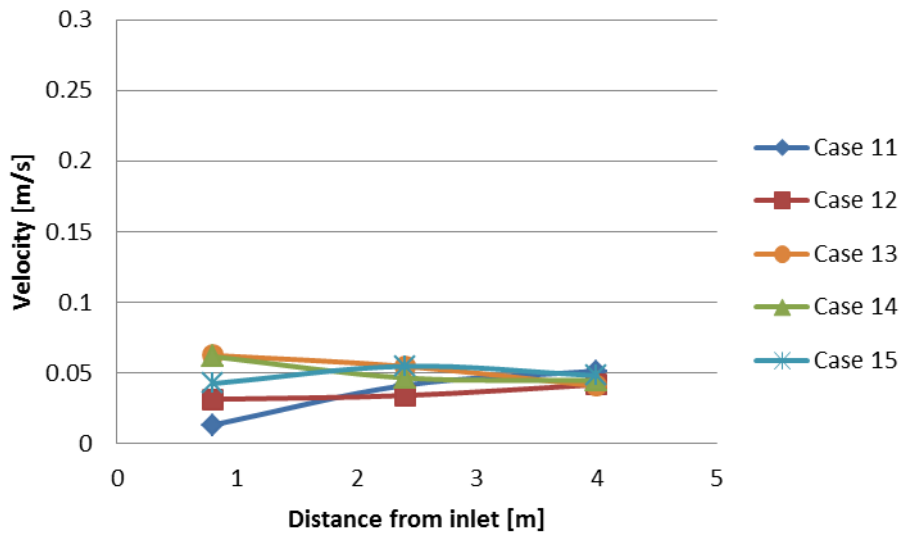


Figure 18 Plenum air temperature distribution and surface temperature distribution of diffuse ceiling

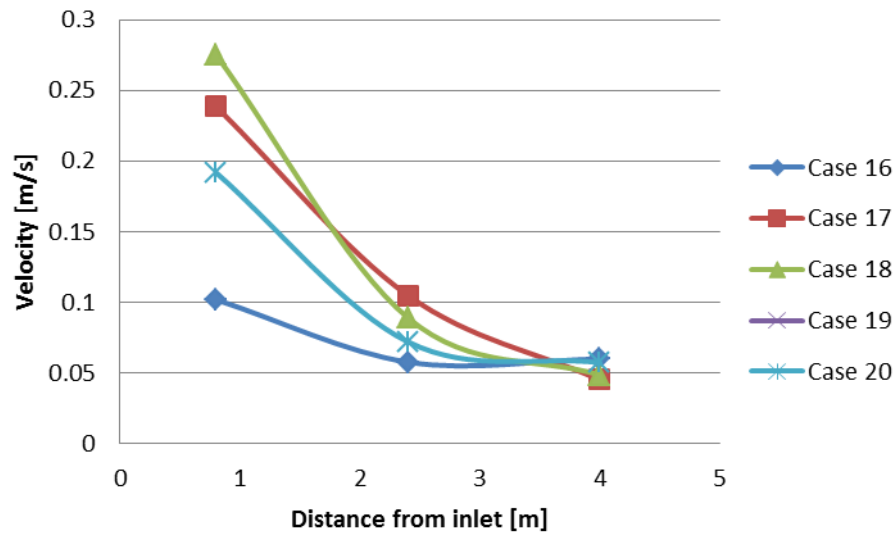
3.6.2 Air velocity distribution in the plenum

Air velocities in the plenum are measured at three locations along the length of plenum as shown in Figure 19.

As illustrated in Figure 19, the velocity distribution in the plenum shows different trend between the cases with air change rate of 2 and those with air change rate of 4. In the cases with low air flow rate, the velocity shows a uniform distribution across the plenum, where the average velocity is around 0.05 m/s. To the contrary, there is a dramatic decrease of velocity as function of distance to plenum inlet in high flow rate cases. In addition, a large velocity difference is found in the position closed. The highest velocity is observed in Case 18 (0.275 m/s), followed by Case 17 (0.238 m/s) and the lowest velocity is in Case 16 (0.102 m/s). This can be explained by the temperature of supply air. when the air is very cold. it will drop down to the ceiling surface and then go through the diffuse ceiling into the occupied zone or travel further in the plenum by attaching the diffuse ceiling surface. Because the anemometers are standing in the plenum, which measure the velocity at 0.15 m height above diffuse ceiling surface, then the air flow attaches diffuse ceiling surface or slab surface can't be measured. The reason for the relative low velocity for Case 20 is that the warm air floats to the upper part of plenum, which can't be measured by anemometer as well.



(a) ACH=2



(b) ACH=4

Figure 19 Air velocities in the plenum

3.6.3 Pressure drop through diffuse ceiling

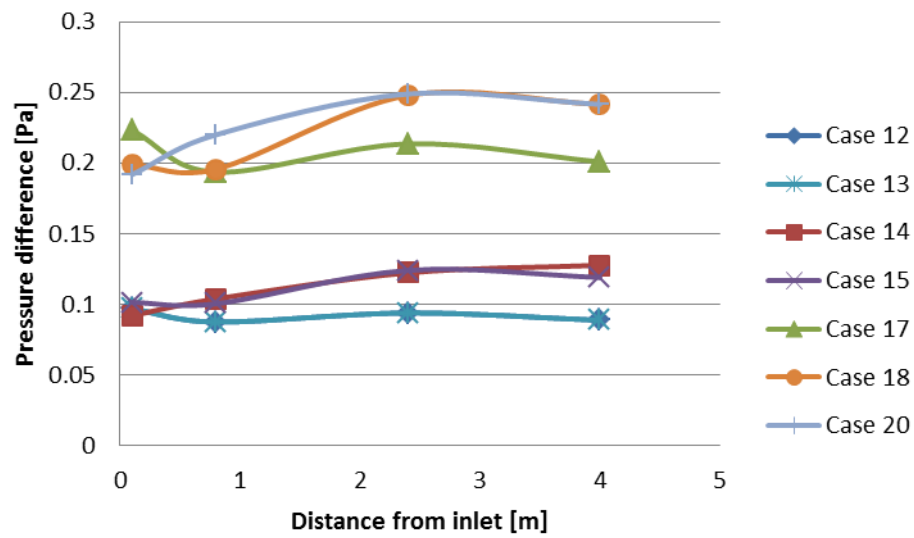


Figure 20 Pressure drop as function of distance to inlet

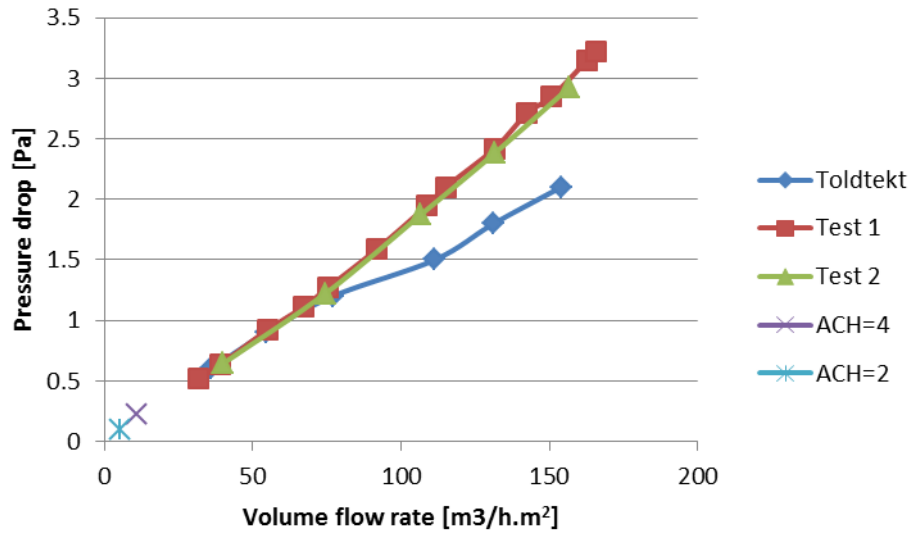


Figure 21 Pressure drop through single acoustic ceiling panel and through entire diffuse ceiling area

Figure 21 illustrate the pressure drop of the single acoustic panel together with the pressure drop of the entire diffuse ceiling. 4 pressure sensors are located in the center line of plenum, and two pressure sensors are positioned in the test room. The pressure drop across diffuse ceiling is approximately 0.1 to 0.12 Pa when air change rate is 2 and 0.2 to 0.25 pa when air change rate increases to 4. From the linearity tendency we conclude that the flow though the diffuse ceiling is laminar.

4. Conclusion

Performance of an integrated system combining diffuse ceiling supply with thermally activated building construction is investigated in this study. Experiments are carried out to examine thermal comfort and energy performance of an office room with integrated system under 20 cases. with different weather condition, internal heat load, activated mode of TABS and with or without diffuse ceiling.

For the cases without diffuse ceiling, the measurements indicate that a linear relation between TABS cooling capacity and log mean temperature difference between operative and fluid. While, for the cases with diffuse ceiling, a power function between cooling capacity and temperature difference is shown. It is clear to see that cooling capacity of TABS is reduced by installing diffuse ceiling, since the direct radiation heat exchange between TABS and other room surfaces is impeded by diffuse ceiling which also changes the air flow patterns. On the contrary, in the heating cases, the heating capacity of TABS is increased with the help of diffuse ceiling. This is because the cold air in the plenum has directly contact with warm TABS surface which significantly enhance the convection heat exchange.

When there is no ceiling panel, the U-value of TABS cooling is almost constant. However, when there exists ceiling panel, this U-value is largely influenced by the plenum and changes with the thermal behavior of this plenum. With the decrease of water temperature the effectiveness of TABS cooling without ceiling panel decreases. But for TABS with ceiling panel, this effectiveness almost keeps constant but very low. This is because the ceiling panel actually increases the thermal resistance between TABS and the room, and lower water temperatures have to be used for supplying the sufficient cooling capacity.

From the view of thermal comfort, diffuse ceiling play a beneficial role on improving thermal comfort in the occupied zone. A good mixing of room air can be achieved with the help of diffuse ceiling. Thus, vertical temperature gradient can be significantly reduced in the occupied zone, especially in the cold climate. Secondly, high draught risk is found at the floor level closed to façade in the cases without diffuse ceiling. This is because that cold air immediately drops to the floor due to gravity. This problem is solved by diffuse ceiling, which adds a resistance layer to prevent cold air directly fall into the occupied zone and provides a pre-heating effect in the plenum. In the case without diffuse ceiling, there is a risk of discomfort by radiant asymmetry when actives TABS in heating mode. Because occupants are more sensitive to warm ceiling then cool one. However, the radiant asymmetry is significantly reduced by diffuse ceiling since the TABS is encapsulated by it and the radiation effect is restrained.

Finally, the air temperature in the plenum and surface temperature of diffuse ceiling point out that the air in the plenum is not perfectly mixing and the air is not evenly distributed through the entire ceiling area, which is as a function of distance to the inlet. However, this does not lead to a thermal comfort issue in the occupied zone. On the other hand, no clear radiation cooling potential of diffuse ceiling is observed in this study, which is due to the low conductivity of diffuse ceiling material. Therefore, there could have future study of the radiation cooling of diffuse ceiling by using other diffuse ceiling types.

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